# POWER

Vol. 38 NEW YORK, SEPTEMBER 16, 1913 No. 12

The Heights by great men reached and kept
Were not attained by sudden flight
But they, while their companions slept
Were toiling upwards in the night
Longfellow
Suggested by
Elton W. Miller
Pittsburg

THE MAN WHO WILL SUCCEED

# Coal Products Company's Power Plant

By S. G. Artingstall, Jr.

*SYNOPSIS—The main features of this plant are found in the type of engines installed and the method of piping the units. Two live-steam headers run parallel with the boiler room. The size of the headers and branch pipes from the boilers are such as to prevent loss of pressure due to high steam velocity.*

The Coal Products Co., Joliet, Ill., has recently put into operation, near that city, a new plant consisting of 35 Koppers improved byproduct coke ovens and other apparatus, having a daily capacity of 450 tons of foundry coke, 3,550,000 cu.ft. of illuminating gas, and the usual quantities of tar and ammonium sulphate.

A bird's-eye view of the plant, taken during the course of erection, is shown in Fig. 1. At the left are the office and byproducts buildings; the double building in the center is the power house, and in front of it the machine shop. The taller chimney is for the coke ovens, which at the time of photographing were not completed; the inclined conveyors carry coal and coke to their repective crusher and storage buildings. At the right is the crane for handling the emergency supply of coal,

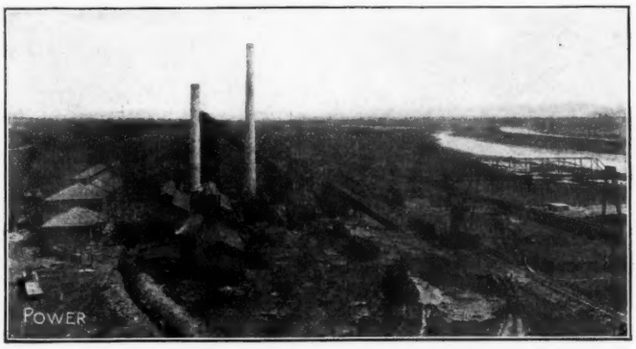



Fig. 1. Birdseye View of the Plant

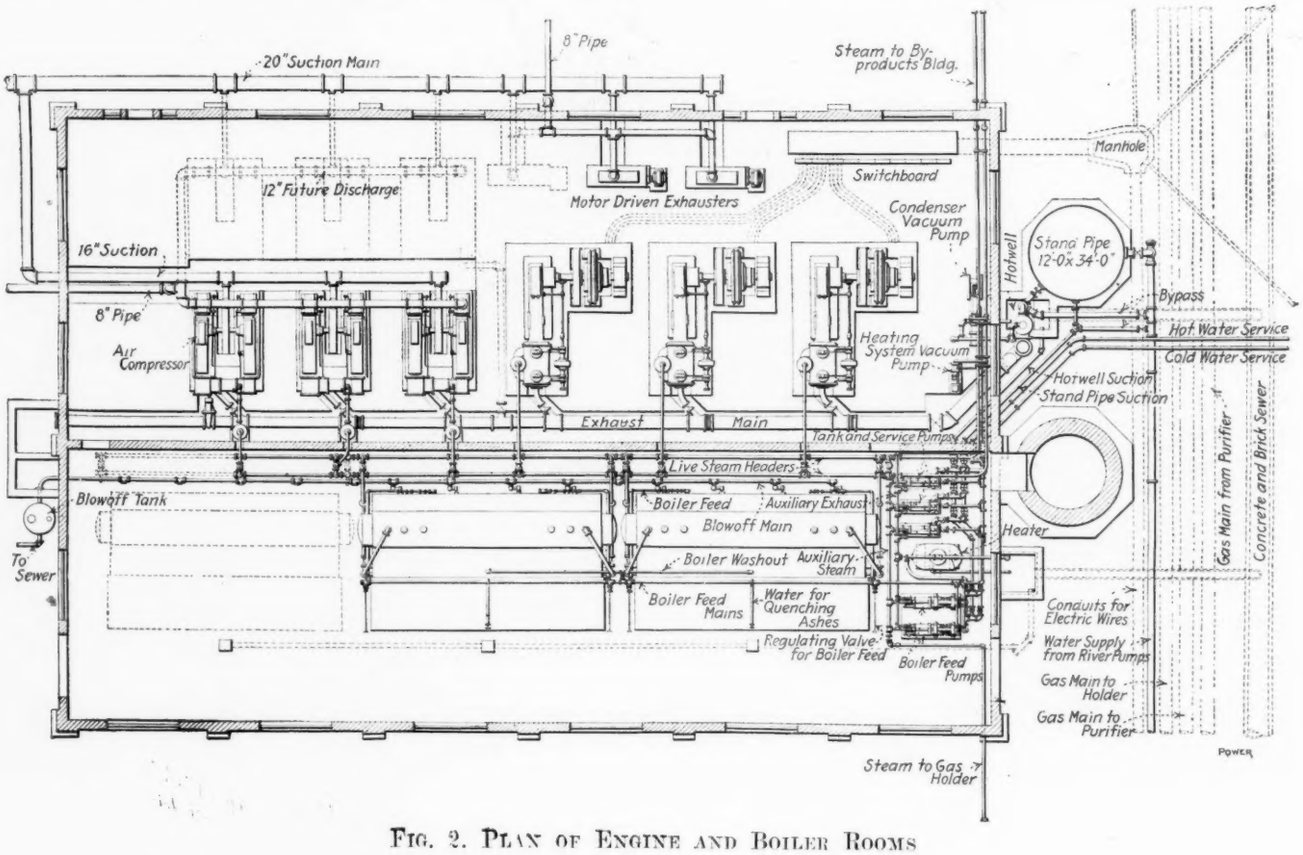



Fig. 2. Plan of Engine and Boiler Rooms

and in the distance is seen the Desplaines River and the Chicago Drainage Canal.

In a plant of this character, it is absolutely essential that an uninterrupted supply of water and electric current be maintained, and although the equipment in the power house is more or less standard, some interesting features are brought out in the design of the piping systems, which are made in duplicate or as interchangeable as possible, so that a complete shutdown will not be necessary in order to make repairs.

## Building

The building, comprising the boiler and engine rooms, a plan of which is shown in Fig. 2, is 78x120-ft. inside dimensions. It is constructed of red pavers over common bricks, which are left rough in the boiler room and plastered in the engine room—the latter being painted green to a height of 10 ft. above the floor, and the balance calcimined white. The roof has steel trusses and purlins, on which is laid a covering of cement tiles. The rock

comes to the surface of the ground at this point, therefore no basement is provided, both floors being placed on the same level—as shown in Fig. 3. All underfoot piping is placed in trenches covered with removable checkered steel floor plates.

### Boiler Room

The boiler-room equipment consists of four vertical water-tube boilers, designed for 160-lb. working pressure, rated at 305 hp. on the basis of 10 sq.ft. of heating surface per horsepower; they are equipped with superheaters to give 125 deg. of superheat. The settings are special, in that the dutch-oven furnaces are extended to give 88 sq.ft. of grate surface. The grates are the pin-hole type with $\frac{5}{16}$-in. holes, the fuel used being coke breeze, hand fired.

The boilers are arranged in two batteries of two units each, space being left for one future battery. An 8x175-ft. radial brick chimney furnishes the draft, aided by a 20-in. turbine blower for each boiler. The blowers are set in the outer casing of the boilers and deliver air through the hollow bridge-wall to openings under the grates, the supply being regulated by dampers operated from the front of the furnaces. The speed of the blowers is governed by a damper regulator, but the dampers in the uptakes to the smoke breeching are adjusted by hand.

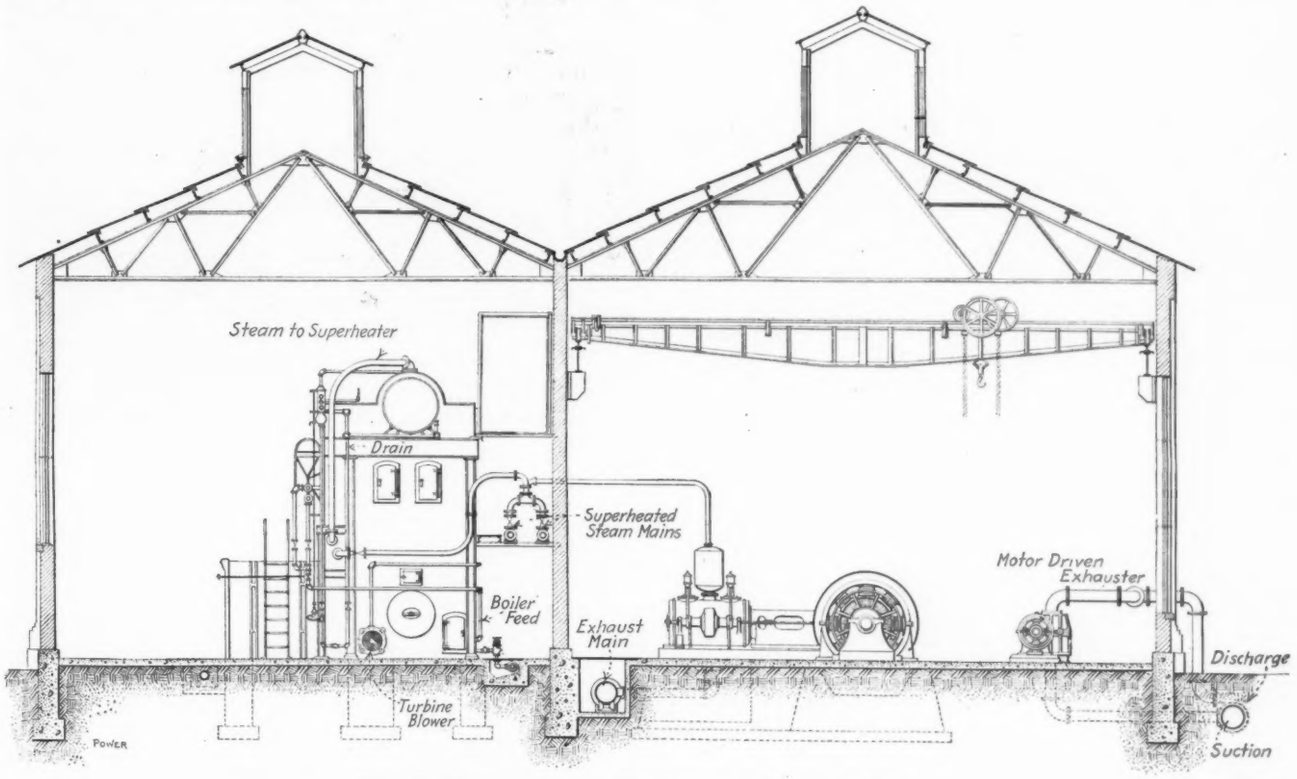



Fig. 3. Elevation of Boiler and Engine Rooms

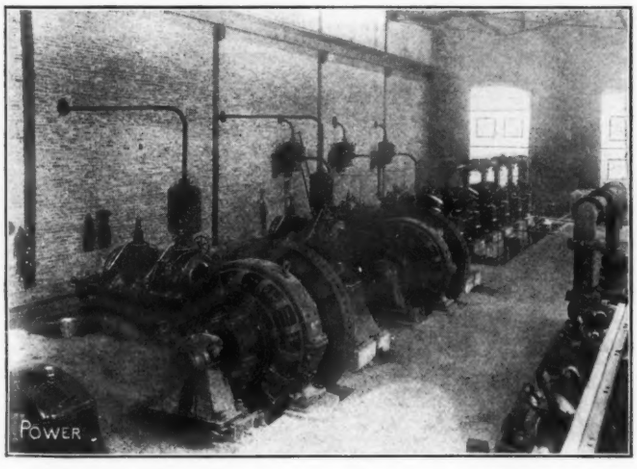



Fig. 4. Engine Room

The boiler feed, tank and service pumps and the heater are arranged in a row at the end of the boiler room, as indicated in Fig. 2.

### Engine Room

In the engine room, Fig. 4, there are three 19x21-in. Lentz engines, directly connected to three 250-kw., 250-volt D.-C. generators.

The power is distributed through a 10-panel marble switchboard, Fig. 5, thence by an underground system to the various buildings about the plant.

Three cross-compound, single-stage gas compressors compress the gas to 40 lb. per sq.in. and deliver it to an 8-in. main leading to Aurora and the more distant towns.

Two rotary blowers of 4.2 cu.ft. per revolution capacity furnish gas to the low-pressure system for the City of Joliet. One of these blowers is coupled to a 25- and the other to a 7½-hp. 230-volt D.-C. motor, the latter being used as a spare unit.

The dry-vacuum pump for the condenser and the heating-system vacuum pump are also located in the engine room. An 18-in. barometric condenser is situated just outside the building at the end of the main-exhaust line from the engines and compressors.

### Water Piping

All piping connecting with the pumps and heater is arranged to give the maximum flexibility, and at the same time is simple. The two service pumps and the heater pump are interchangeable, and can take water from the standpipe or hotwell and deliver it to the hot- or cold-service system or to the heater, or all the pumps can be bypassed and the heater take its supply directly from the standpipe by gravity.

Both boiler-feed pumps can take water from the standpipe, hotwell or heater; while one is feeding the boilers, the other may be delivering water for washing purposes, there being two mains running parallel to and in front of the boilers for these requirements. A hose connection for each battery and the boiler-feed branches to each boiler are connected to both of these mains.

### Pump House

The house and the pump supplying water to the standpipe are situated about 1000 ft. away. A channel blasted out of the rock connects the suction pit to the main channel of the river. The equipment consists of two motor-driven 8-in. centrifugal pumps, each connected through automatic starters to float switches in the standpipe. One of the floats is situated near the top at the regular water line, and the other about half-way down, so that if the upper one fails to operate, the lower one will cut in the idle motor before the standpipe is empty. A double-throw switch on the panelboard in the pump house reverses the floats, and is thrown once a month so that each motor gets an equal share of the work.

### Live-Steam Piping

The system of live-steam piping consists of two headers, running parallel to and in back of the boilers to the end of the boiler room, then turning at right angles and continuing to the byproducts building, as shown in Fig. 2. Connections to these headers are made from the boilers and to the engines, Fig. 3. A U-bend in each header, about in the center, takes care of the expansion, which is divided up by means of anchors; a walk on a level with the headers gives a means of easy access to all the valves.

Full-weight pipe is used with semi-steel screwed flanges, semi-steel fittings and valves, the latter having nickel seats, disk rings and nickel-steel rising stems. A double-thick covering of 85 per cent. magnesia insulates the entire system.

In designing the live-steam piping, advantage has been taken of past experience, which has shown that the usual rule of a velocity of 6000 ft. per minute would not apply for superheated steam, on account of the excessive loss in temperature at the throttle; on the other hand, it is well known that as the velocity increases the pressure decreases, until the advantage gained in holding the superheat is more than offset by the resulting drop. Therefore, a medium has been struck between these two extremes, and 4-in. pipes for the headers and branches from the boilers, and 3-in. pipes to the receivers were installed. The receivers are of ample capacity to prevent fluctuations and wire drawing, and are heavily lagged with 85-per cent. magnesia. Receivers for the engines are placed directly on the throttle and those for the compressors on the wall.

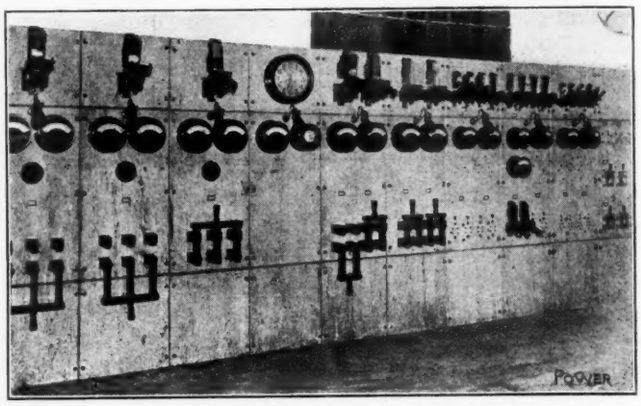



Fig. 5. Switchboard

It was considered more economical, from the standpoint of upkeep, to operate the compressors with less superheat than that used for the Lentz engines, which are made especially for high superheat. The receivers for the compressors are, therefore, placed on the wall and some distance away and connected to the throttle with a 5-in. pipe to reduce the velocity of the steam. Provision is also made to keep a stream of water circulating through these receivers to kill the superheat, if it is found necessary.

This system has fulfilled all expectations, as shown by actual operating conditions; the drop in pressure is slight, and very little superheat is lost. The receivers, acting as reservoirs, counteract any disadvantage due to the smallness of the piping, and at the same time the total radiating surface is reduced fully 30 per cent. Another advantage is the cost; the smaller fittings, valves, etc., creating quite a saving.

PRINCIPAL EQUIPMENT OF THE COAL PRODUCTS CO.'S POWER PLANT

| | Equipment | Type | Purpose | Kw. | Volts | Amp. | R.p.m. | Size | Pres. | Hp. | Manufacturer |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 4 | Boilers | Vert. W.T. | Steam Generation | .... | .... | .... | .... | .... | 160 | 305 | Erie City Iron Works |
| 4 | Superheaters | Foster | 125 degrees S.H. | .... | .... | .... | .... | .... | 160 | .. | Power Specialty Co. |
| .. | Grates | Pin-hole | Furnace | .... | .... | .... | .... | .... | .... | .. | Erie City Iron Works. |
| 4 | Blowers | Turbine | Draft | .... | .... | .... | Variable | 20 in. | 160 | .. | L. J. Wing Mfg. Co. |
| 1 | Regulator | Spencer | Blowers | .... | .... | .... | .... | .... | .... | .. | Thompson, Richards & Co |
| 1 | Chimney | Radial Brick | Draft | .... | .... | .... | .... | 8x175 ft. | .... | .. | Heinicke Chimney Co. |
| 3 | Engines | Lentz, Simple | Main Units | .... | .... | .... | 200 | 19x21 in. | 150 | 325 | Erie City Iron Works. |
| 3 | Generators | Direct Current | Main Units | 250 | 250 | 1000 | 200 | .... | .... | .. | Westinghouse Elec. & Mfg. Co. |
| 3 | Compressors | Single-Stage Compound | Gas | .... | .... | .... | Variable | 14x24x16 18x16 in. | 150 | .. | Ingersoll-Rand Co. |
| 2 | Blowers | Rotary | Gas | .... | .... | .... | Variable | 4.2 cu.in. | .... | .. | Connersville Blower Co. |
| 1 | Condenser | Barometric | Main Units | .... | .... | .... | .... | 18 in | .... | .. | Alberger Condenser Co. |
| 1 | Pump | Dry Vacuum | Condenser | .... | .... | .... | .... | .... | .... | .. | Alberger Condenser Co. |
| 1 | Pump | Vacuum | Heating System | .... | .... | .... | .... | .... | .... | .. | Consolidated Engineering Co. |
| 2 | Pumps | Duplex O.C.P. | Boiler Feed | .... | .... | .... | .... | 10–6–10 in. | .... | .. | Canton Hughes Pump Co |
| 3 | Pumps | Duplex | Service & Heating | .... | .... | .... | .... | 8–8½–10 in. | .... | .. | Canton Hughes Pump Co |
| 1 | Heater | Open | Boiler Feed | .... | .... | .... | .... | 4 | .... | .. | Stillwell-Bierce & Smith-Vaile Co. |
| 3 | Governors | Fisher | Service Pumps | .... | .... | .... | .... | ¾ in. | .... | .. | Fisher Governor Co. |
| 1 | Switchboard | Marble | Electrical Distribution | .... | .... | .... | .... | 10 pan | .... | .. | J. Lang Electric Co. |

### Exhaust Piping

Exhaust steam from the prime movers is carried in a 16-in. main at the compressors, increasing to 18 in. at the engines, and is made of ¼-in. material with vanstone joints. Expansion is taken up in corrugated-copper expansion joints.

The condenser at the end of the exhaust main gives a 26-in. vacuum, with the aid of a dry-vacuum pump. The condenser draws its injection water directly from the standpipe, by means of its own vacuum.

Exhaust steam from the auxiliaries is carried to the heater, which raises the boiler-feed water to about 200 deg. F.

The plant, as a whole, has proved very economical as to operation and maintenance, and the live-steam piping is rather exceptional when compared with the piping usually found in the average power house.

# Feed-Water Heating for Locomotives*

*SYNOPSIS—Feed-water heating is first considered as to its beneficial effects on the boiler and the requirements of a good heater are set forth. Finally, comparative results are shown for locomotive practice of the savings from feed-water heating when using saturated steam, moderate superheat and high superheat.*

One practical advantage occurring from the use of a feed-water heater is the greater protection given the boiler by reason of the lesser temperature changes. While steam at 170 lb. gage has a temperature of about 375 deg. F., that of the water in the tender usually rises but little above the temperature of the water taken in, and in winter the water in the tender is cooled down to near the freezing point. Of course, feeding is not done at this low temperature, for with the injector the water is warmed perhaps 72 to 90 deg. Nevertheless, in winter there are to be found in the boiler all temperatures between 120 and 375 deg. F. These temperature changes give rise to distortions resulting in leakage at the rivet joints and at the tube-sheet. When feeding with water heated to 265 deg. the temperature in the boiler remains more uniform. In this case, between the points of lowest and highest temperatures there obtains a difference of, say, 375 — 265 = 110 deg. against 375 — 122 = 253 deg. without a heater. A further safeguard, introduced by the use of a feed-water heater, is that the sediment and scale are no longer deposited alone in the boiler but to a large extent in the heater where they are less injurious.

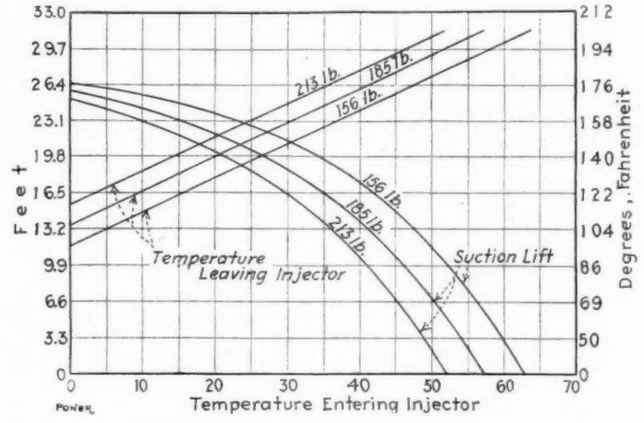



Fig. 1. Comparison between Temperature of Feed Water Entering and Leaving Injector; Also Theoretical Suction Lift

From the foregoing, two essentials may be advanced as necessary in the design of the heater. First, it should permit of the deposit of the sediment at some dead point that it may not be carried into the boiler where it might prove troublesome at the check valve, or injure the injector or the pump if these latter be located between the heater and the boiler, and the construction should be such as will readily admit of cleaning. However, scale which may develop on the heater shell is harmless, as the metal is not exposed to the action of the fire. From the standpoint of easy cleaning preference is to be given to that type of heater where the water flows through straight tubes rather than to that where the tubes are surrounded by the water, and also to comparatively large tubes rather than small ones.

In addition to the mineral matter separated from the water by the heater the dissolved air and carbonic acid are expelled; this also operates strongly toward preservation of the boiler surfaces, and in locomotives provided with feed-water heaters pitting and the formation of rust are decidedly less marked.

Preheating the feed water avoids the sudden pressure drop in the boiler on starting the injector. Along with the advantages of simplicity, light weight and small space requirement the injector has the disadvantage that in practice its delivery volume can be but slightly varied. On grounds of safety it is proportioned to meet the maximum demand possible and is thus only intermittently in use, whence its steam consumption is large as its efficiency is only 5 or 6 per cent. Added to the high steam consumption under forced duty, 8 to 9 per cent. of that of the engine, the introduction into the boiler of comparatively low-temperature feed water induces pressure drops which are very annoying under operating conditions. If the injector forces the feed through a heater before entering the boiler, or if it be drawn through one by a pump, and raised to a temperature of 212 to 265 deg. the pressure drops in the boiler while feeding are materially lessened.

The theoretical suction lifts of the injector and the discharge temperatures due to the suction temperatures are shown in Fig. 1 for boiler pressures of 156, 184 and 213 lb. The results attained in practice do not fully agree with the values given, owing to incomplete condensation in the injector and heat and friction losses. The actual temperatures of delivery lie somewhat higher; the suction heights somewhat lower. In any case the temperature of the discharge is certainly below 212 deg. F. With a feed-water temperature of 50 deg that of the dis-

*Translated in abstract by Wm. F. Monaghan from a recent contribution to "Zeitschrift des Vereines Deutscher Ingenieure," by Ludwig Schneider.

charge from the injector, under normal conditions, is about 140 deg.

The ordinary injector refuses to lift water at a temperature of 104 deg. F. The manipulation of the so called hot-water injector, which operates with a suction temperature of even 212 deg., is too complex for use so far on locomotives. Steam pumps used for feeding deliver, under usual boiler pressures, from 100 to 150 lb.

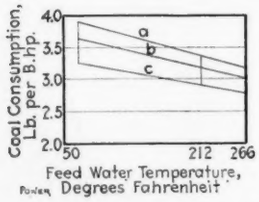



FIG. 2. COAL CONSUMPTION OF SIMPLE ENGINE AS DEPENDENT ON FEED-WATER TEMPERATURE

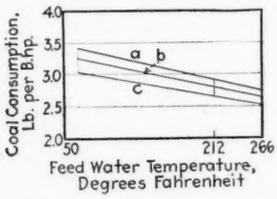



FIG. 3. COAL CONSUMPTION OF COMPOUND ENGINE AS DEPENDENT ON FEED-WATER TEMPERATURE

of water per pound of steam consumed as against only 15 lb. by an injector. In addition the exhaust from the pump can be utilized for heating the feed water, and its delivery volume can be simply and positively varied within wide limits.

### ECONOMY AND THEORY OF PREHEATING

The process of heat transfer from the hot gases to the boiler contents resolves into two divisions, or with compound engines into three. In the first place the water must be heated to the evaporating temperature. This, with a boiler pressure of about 213 lb., is represented by 392 deg. To heat the feed water from 50 to 393 deg. there must be supplied to each pound approximately 343 B.t.u. The latent heat of evaporation under these conditions amounts to 833.5 B.t.u. The total heat expenditure for the generation of steam at a pressure of about

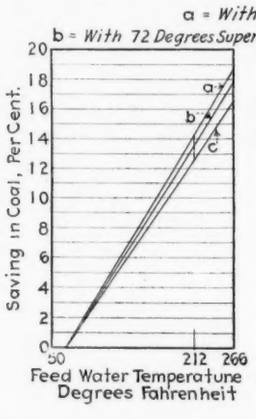



FIG. 4. SAVING IN COAL THROUGH HEATING FEED WATER WITH SIMPLE ENGINE

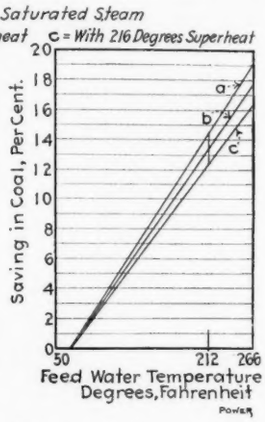



FIG. 5. SAVING IN COAL THROUGH HEATING FEED WATER WITH COMPOUND ENGINE

213 lb. from 1 lb. of water at a temperature of 50 deg. is thus

$$343 + 833.5 = 1176.5$$

If the steam be superheated to 600 deg. there will be the further requirement expressed by the product of the temperature increase and the average specific heat; or for the foregoing,

$$(600 - 393) \times 0.552 = 114.2 \text{ } B.t.u.$$

Under this assumption the total heat absorption amounts to

$$1176.5 + 114.2 = 1290.7 \text{ } B.t.u.$$

Now if the feed water be heated first by exhaust steam to 212 deg., there will result a saving of 162 B.t.u. The heat transfer within the boiler is lowered, in the case of saturated steam, by 13.8 per cent., and in the case of superheated steam by 12.5 per cent., and assuming that preheating of the feed water does not change the efficiency of the boiler these values represent at the same time the saving in fuel. As a matter of fact, due to the maintained cleanliness of the boiler, resulting from the use of a heater, its efficiency is raised and the percentages given will even be exceeded. The values presented in

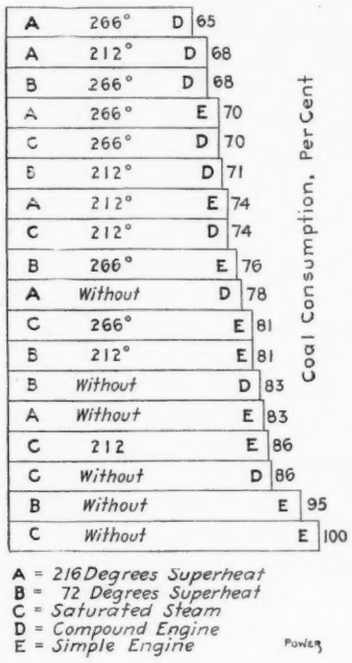

A 266° D 65
A 212° D 68
B 266° D 68
A 266° E 70
C 266° D 70
B 212° D 71
A 212° E 74
C 212° D 74
B 266° E 76
A Without D 78
C 266° E 81
B 212° E 81
B Without D 83
A Without E 83
C 212 E 86
C Without D 86
B Without E 95
C Without E 100
Coal Consumption, Per Cent
A = 216 Degrees Superheat
B = 72 Degrees Superheat
C = Saturated Steam
D = Compound Engine
E = Simple Engine
Power

FIG. 6

Figs. 2 to 6 for constant boiler efficiency are, however, calculated without reference to this favorable condition. Also for the assumed boiler efficiency and steam consumption it was sought to maintain coal-consumption values easily reached in practice, so that there is no room for distrust of the calculated results. The heat value of the coal is estimated at 12,625 B.t.u., and the boiler efficiencies, inclusive of the superheaters for (a) as 70 per cent., for (b) as 70 per cent., and for (c) as 67 per cent.

The simple engine, with saturated and with superheated steam, may be first considered. When operating without preheating the feed water, superheating the steam results in a coal saving of 15.5 per cent.; with coincident preheating and superheating this saving drops in a remarkable way to 13 per cent., and with the feed water preheated to 266 deg. F. the saving amounts to only 12.5 per cent. Thus the influence of feed-water heating on the utility of superheating, insofar as it is expressed in coal consumption, is similar to that of multiple expansion; that is, it lessens the relative economies so little that there is no question of the virtue of superheating.

A like consideration of the compound engine shows that high superheating without feed-water heating yields a saving of coal of 10.5 per cent. as opposed to saturated

steam. Heating the feed water to 212 deg. F. shows a saving of 8 per cent. and with preheating to 266 deg. a saving of 7.5 per cent. The law thus holds good for both simple and compound engines.

Based on the coal consumptions, as of Figs. 2 and 3, a graphic series may be arranged as in Fig. 6 for simple and compound locomotives using saturated or highly superheated steam or with a steam dryer. These show 18 different conditions of operation, from the simple engine using saturated steam and without feed-water heating

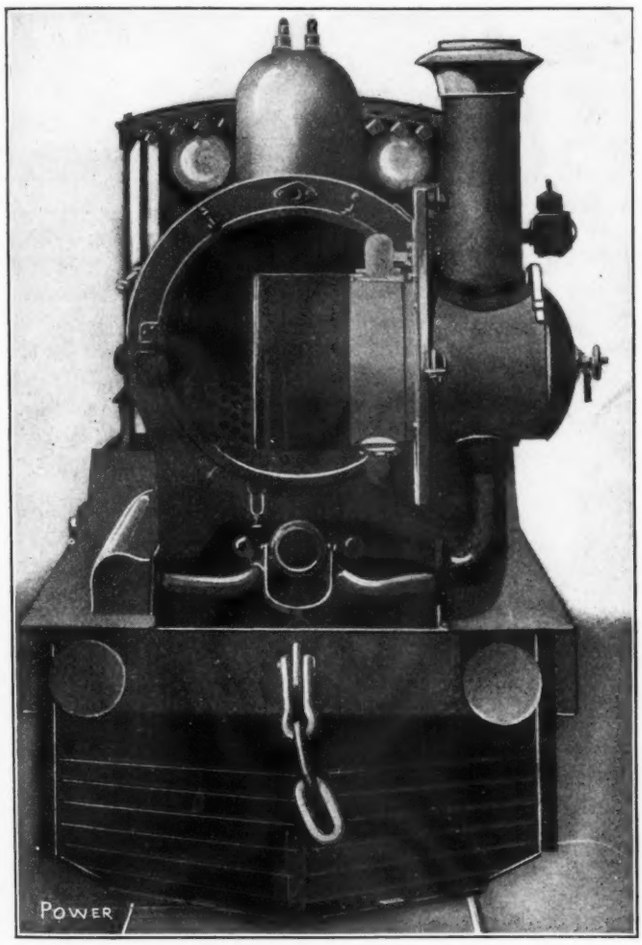



FIG. 7. LOCOMOTIVE WITH FEED HEATER IN UPTAKE

and burning 3.85 lb. of coal per horsepower-hour measured at the rail, to the compound engine using steam of high superheat and with preheated feed water and consuming 2.508 lb. per hp.-hr., a ratio of 100 to 65.

A feed-water temperature of about 212 deg. F. can be reached by condensing in a suitable heater the exhaust from the air and boiler-feed pumps and using in addition a part of the main exhaust. To heat a given quantity (F lb.) of feed water from 50 to 212 deg. there will be needed, of dry exhaust steam, an amount represented by the quotient of the latent heat of steam at atmospheric pressure into the temperature range of heating, or

$$F\,\frac{162}{970.4} = \frac{F}{6}$$

In other words, for preheating, about one-sixth of the exhaust is necessary. The air-pump exhaust may be estimated at 3 per cent. of that from the engine cylinders per hour; the boiler-feed pumps, with an hourly feed delivery of $F$ pounds, will furnish in pounds of exhaust steam about

$$\frac{F}{100}$$

then if these quantities be supplied to the heater there will be a further requirement of about one-eighth of the main exhaust. In view of the moisture in the steam this figure will be somewhat increased so that it may be taken as one-seventh of that exhaust when operating with superheated steam and one-sixth with saturated steam. In general it will be of advantage to regulate this quantity of steam to suit the amount of feed at any given time in order that the blast may not be unnecessarily depreciated.

The heat-transferring surface of the heater may be calculated by means of the formula

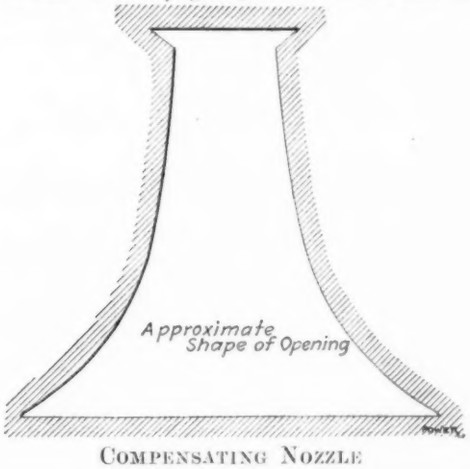
$$S = \frac{Q}{ct}$$

where $S$ is the heating surface in square feet; $Q$ the quantity of heat to be transferred per hour in B.t.u., $c$ the coefficient of heat transference, and $t$ the average temperature difference of the steam. The area thus obtained is to be increased by a certain amount to allow for losses due to incrustation and to outward radiation; an amount dependent on the hardness of the water and the design of the heater. This method gives a heating surface for the feed-water heater of about $^1/_{12}$ to $^1/_{18}$ of that of the boiler.

The author refers to feed-water heaters utilizing the waste gases and used alone or in combination with those where heat is absorbed from the exhaust steam, and includes a number of other curves and formulas and reproductions of interesting drawings and photographs of domestic and foreign practice, notably the Gaines method used on the Central of Georgia R.R. in this country, employing both of the types of heaters referred to and a reciprocating feed pump, and the odd designs in use on the Egyptian State Railroads, one of which is shown in Fig. 7.

## Compensating Device for Water Measurement

There are a number of water meters on the market, some of which are very good, when cold water only has




COMPENSATING NOZZLE

to be dealt with, but the number of meters which can accurately integrate and record the flow of hot water such as boiler feed in the hot-well tank in the power

house is limited. Of those which give the best results, probably the most successful form is that which depends for its measurement on the flow of water over a weir or notch.

Considering this flow, the law connecting the rate at which the water is discharged over the weir and the depth of water over the weir is in the nature of

$$R = K S \sqrt{D}$$

where

$R$ is the rate of discharge at any time;

$D$ is the depth of water over the weir;

$S$ is the submerged area of the weir;

$K$ is a constant which includes the coefficient of discharge.

It will be seen that this is a complex formula, in which one of the variables follows the square-root law. Where each rate of discharge has to be calculated independently, the expression is sufficiently simple to allow the calculation to be carried out easily, but for an integrating or recording instrument in which measurements have to be taken continuously and automatically, the next step in the simplification of the instrument is to eliminate, if possible, the square root in the foregoing expression.

In the instrument constructed by Alexander Wright & Co., London, a device has been applied which consists in making the orifice of the weir somewhat in the form illustrated herewith, in which the breadth of the weir is made inversely proportional to the square root of the corresponding depth at any level, with the result that the depth becomes directly proportional to the flow. The sides of the notch are obviously a logarithmic curve. With this notch it is possible, by means of a float and connection to a recording pen on a rotating drum to draw curves showing the rate of discharge at any instant on an evenly divided vertical scale, and a curve of this description is obviously much more simple of measurement than one in which the vertical heights follow the square-root law.

A second device is incorporated in the water meter as regards the method of integrating. A ground-glass plate is mounted in a horizontal position in gear with the clock which drives the chart drum. This revolves on a vertical axis at a uniform rate. The float rod which is moved upward or downward in accordance with the depth of water in the weir, has connected to it a small index and driving wheel which is moved outward from the center of the glass as the float rises. The wheel is rotated by friction with the ground-glass disk, and obviously the further out that it proceeds from the center of the disk the more rapid is its travel in a given time. In this way the number of rotations of the index wheel in a given time is varied automatically, and through a very fine series of gradations according to the variations of the depth of water in the weir. The action is accurate, immediate and simple, and in this way the integration is readily obtained.

It will be seen therefore that this "Precision" water meter is capable of being adapted to a large range of water measurements, the rate of flow being indicated on the chart and the total amount of water being shown on the index in plain figures.

❋ ❋

# Distribution of Heat in Steam Boilers*

*SYNOPSIS—A study of the losses encountered in the steam boiler and some test results, showing how the boiler efficiency was increased.*

❋

In accordance with the recent recommendations of the power-tests committee of the American Society of Mechanical Engineers, a balance accounting for the distribution of heat in a pound of dry coal fired consists of the following items: 1. Heat absorbed by the boiler. 2. Loss due to evaporating and superheating moisture in coal. 3. Loss due to evaporating and superheating moisture formed by the combustion of hydrogen and distillation of oxygen-hydrogen compounds in the coal. 4. Loss due to heat carried away by dry flue gases leaving the boiler. 5. Loss due to heat escaping through the formation of carbon monoxide (CO). 6. Loss due to combustible removed from ashpit and from grates during cleaning. 7. Loss due to superheating moisture in the air used for combustion. 8. Loss due to unconsumed hydrogen and hydrocarbons, to radiation and unaccounted for.

## Heat Carried Away by Flue Gases

Of the items enumerated, 4, 5 and 6 generally constitute about 75 per cent. of the heat lost, and are the variable factors which have a direct effect on the percentage of heat imparted to the water. The loss which is generally the largest and in the majority of cases most easily reduced is the heat carried away by the dry flue gases escaping from the boiler. Under this heading are included the sensible heat in the carbon dioxide and carbon monoxide, if any, and the heat in the excess air. The loss depends upon the weight of gases per pound of carbon burned together with the temperature at which the gases leave the boiler. The data which are required for this computation are the temperature of the flue gases, temperature of the boiler room, composition of the flue gases, the heat value of the coal and the carbon content of the fuel and ashpit refuse.

Several tests conducted in a plant in which the loss of heat in the flue gases was originally high are submitted in Tests A of Table 1. The plant in question consisted of hand-fired horizontal return-tubular boilers equipped with plain grates upon which Pennsylvania semibituminous coal, containing about 9 per cent. of ash and 22 per cent. of volatile matter, was burned. The fires were carried at a thickness not exceeding 7 in. and were uneven, thereby permitting a large air excess. Increasing the thickness of the fire and altering the methods of firing resulted in a reduction of the loss of heat due to excess air from 36 to 12 per cent., with a resultant increase of about 20 per cent. in the boiler efficiency.

## Loss from Carbon Monoxide

The loss of heat due to the incomplete combustion of carbon to carbon monoxide is usually an important fac-

*Presented by Perry Barker, fuel engineer of Arthur D. Little, Inc., at the fifth semi-annual meeting of the American Institute of Chemical Engineers, Boston, Mass.

tor. While the presence of carbon monoxide shows that some heat is lost through incomplete combustion, a small percentage of this constituent generally indicates that the air excess is reduced as low as practicable. A small amount of smoke, as evidence of the presence of carbon monoxide, is a more convincing indication of economical furnace operation than a smokeless stack which may be emitting several hundred per cent. of excess air.

The extent of this loss is dependent upon the percentages of carbon monoxide and carbon dioxide as determined by the flue-gas analysis, the heat value of the coal and the percentage of carbon in the coal fired and in the ashpit refuse.

Results of several tests in a plant where the loss due to unburned gases, as indicated by the presence of carbon monoxide was particularly high, are shown in Tests B of Table 1. The boiler equipment consisted of hand-fired vertical tubular boilers having shaking grates. Coal from the Georges Creek field, containing approximately 18 per cent. of volatile matter was burned. The high loss due to incomplete combustion was not attributed to the character of the coal, although it had been proved that the furnace space in the boilers was not sufficient to burn coals containing higher percentages of volatile matter without the production of excessive amounts of smoke, indicating a large loss, due to incomplete combustion. Aside from the question of furnace design, the primary cause of the high percentages of carbon monoxide was the thickness of the fires. By reducing the thickness from 30 to 12 in. and changing the method of firing, the loss due to this factor was reduced from 14 to 1 per cent., with a corresponding increase in boiler efficiency.

## Ashpit Loss

In the operation of a boiler plant a certain percentage of the fuel which is fed to the furnace drops through the grate and is removed from the ashpit. An additional quantity of unburned coal is drawn from the grates in hand-fired plants when the fires are cleaned. If the coal is fine or noncaking and the grates contain large air spaces, excessive amounts of fuel drop into the ashpits. Certain coals clinker badly under some conditions of operation and, therefore, during the removal of this clinker large amounts of unburned coal are drawn from the fires. Another cause of this loss is the frequent barring or raking of the fires when a fine or noncaking coal is used.

The ashpit loss which also includes the combustible removed during cleaning is determined from the weight of this material, the percentage of combustible which it contains together with the weight of the dry coal fired. For the purpose of this computation the heating value of the combustible is taken as 14,600 B.t.u. per lb. In special cases where complete boiler tests are not made and the weights of coal and ashpit refuse are not obtained, the percentage of the heat in the fuel removed with the ashpit refuse and cleanings can be computed from the analysis of this material and the determination of the percentage of ash in the coal as fired. In calculating this item by the latter method the assumption must be made that the percentage of ash obtained by the proximate analysis of the fuel represents the inert matter in the ash and refuse removed from the ashpits and the furnaces; that is, no account is taken of the ash which is carried off the grates and deposited in the combustion chamber, upon the water-heating surface or lost up the stack. A series of tests which indicate the extent to which this loss can be reduced are given in Tests C of Table 1. This plant was equipped with inclined grate stokers upon which Pittsburgh slack coal was burned. The excessive rate at which the fuel was burned to maintain the required capacity caused deterioration of the grates, which allowed large amounts of unburned coal to fall into the ashpits. By careful attention to grate repairs and methods of operating the stokers the boiler efficiency was increased approximately 10 per cent.

## Loss Due to Hydrogen

The percentage of hydrogen in various coals is not an indication of their relative value when burned for the generation of steam. The essential feature in determining the effect of this constituent in the process of combustion is the form in which it is available in the furnace. Hydrogen which is driven off in the uncombined state and burned by mixing with the oxygen from the air supply is a valuable factor as it develops about four times the heat generated by an equal percentage of carbon. For the purpose of this discussion the theory of the code of the American Society of Mechanical Engineers for the loss due to hydrogen, is accepted. This theory assumes that all hydrogen contained in the fuel, together with an equivalent amount of oxygen in the form of water is raised from boiler-room temperature, evaporated in the furnace and is eventually superheated to the temperature of the escaping gases. The amount of heat lost in this manner depends upon the composition of the fuel, the temperature of the boiler room and the temperature of the escaping gases In addition to the loss due to evaporating and superheating the moisture formed by the combustion of hydrogen and by the distillation of oxyhydrogen compounds, a small loss is incurred by subjecting the moisture in the fuel and in the air used for combustion to these processes.

TABLE I. DECREASE OF LOSSES DUE TO FLUE-GAS TEMPERATURE, CARBON MONOXIDE AND UNBURNED FUEL

| | Tests A | | | Tests B | | | Tests C | | |
|---|---|---|---|---|---|---|---|---|---|
| | Original Conditions | Improvement Effected | | Original Conditions | Improvement Effected | | Original Conditions | Improvement Effected | |
| 1. Loss due to heat carried away by the dry flue gases | 35.9 | 10.0 | 12.4 | 15.3 | 1.1 | 2.1 | 13.1 | 9.1 | 7.9 |
| 2. Loss due to evaporating and superheating moisture in coal and moisture formed by the combustion of hydrogen and distillation of oxyhydrogen compounds | 2.7 | 2.7 | 2.8 | 4.1 | 3.8 | 4.2 | 3.2 | 3.3 | 3.4 |
| 3. Loss due to heat escaping through the formation of carbon monoxide | 0.0 | 0.4 | 1.6 | 9.7 | 15.1 | 10.7 | 22.8 | 19.1 | 18.2 |
| 4. Loss due to combustible removed from the ashpits and from the grate during cleaning | 3.2 | 7.6 | 4.5 | 3.2 | 3.5 | 3.3 | 0.0 | 0.0 | 0.0 |
| Total determined losses | 41.8 | 20.7 | 21.3 | 32.3 | 23.5 | 20.3 | 39.1 | 31.5 | 29.5 |
| 5. Loss due to unconsumed hydrogen and hydrocarbons, to radiation and unaccounted for (assumed constant for all tests) | 8.9 | 8.9 | 8.9 | 11.2 | 11.2 | 11.2 | 7.5 | 7.5 | 7.5 |
| 6. Approximate percentage of heat absorbed by the boiler or boiler efficiency (by difference) | 49.3 | 70.4 | 69.8 | 56.5 | 65.3 | 68.5 | 53.4 | 61.0 | 63.0 |
| Total | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 |

Increasing interest in the economical operation of boiler plants has resulted in marked improvement in efficiency in the majority of the plants where this matter has been given careful attention. An illustration of the distribution of heat in a boiler plant which was operating with high efficiency before tests were made for contemplated improvement is shown in Table 2. This plant consisted of large units of Climax water-tube boilers

TABLE 2. PLANT OPERATING AT HIGH EFFICIENCY

| | Original Conditions | Improvement Effected | |
|---|---|---|---|
| 1. Loss due to evaporating and superheating moisture in coal and moisture formed by the combustion of hydrogen and distillation of oxyhydrogen compounds | 4.0 | 3.8 | 3.8 |
| 2. Loss due to heat carried away by the dry flue gases | 14.4 | 10.1 | 11.8 |
| 3. Loss due to heat escaping through the formation of carbon monoxide | 0.9 | 1.3 | 0.4 |
| 4. Loss due to combustible removed from the ashpits and from the grate during cleaning | 0.6 | 0.8 | 0.9 |
| Total determined losses | 19.9 | 16.0 | 16.9 |
| 5. Loss due to unconsumed hydrogen and hydrocarbons, to radiation and unaccounted for (assumed constant for all tests) | 8.0 | 8.0 | 8.0 |
| 6. Approximate percentage of heat absorbed by the boiler or boiler efficiency (by difference) | 72.1 | 76.0 | 75.1 |
| Total | 100.0 | 100.0 | 100.0 |

equipped with circular grates which were fired through a number of doors. It will be noted from the tabulation that considerable loss, due to the formation of carbon monoxide, occurred, but this waste is more than counterbalanced by the particularly small percentage of combustible removed from the ashpits and furnaces. The amount of combustible in this material averaged about 12 to 15 per cent., representing a fuel loss of from 0.5 to 1 per cent. Although the operating conditions in this plant were excellent, still greater care in the methods of firing effected an increase in boiler efficiency.

## Mexican Boiler Explosion

By T. E. Dutton

The accompanying photographs show the results of a recent boiler explosion in Mexico. There were four 125-hp. fire-tube boilers in a row, and a smaller one behind them. One of the center boilers exploded, demolishing the setting of all the boilers. The exploded boiler was thrown forward about 50 ft. It was made with two sections and the front plate tore away from the rear plate, shearing nearly all of the rivets on the girth seam. The front plate, which failed, ripped along over the fire. It was found badly corroded and extremely thin in places, which was no doubt the cause of the explosion.

One fireman was killed and another injured. The accident happened at night and but few men were about.

Even a superficial inspection would have exposed the state of the boiler, but boiler inspectors are an unknown quantity down here in Mexico.

## Settled Ten Feet but Kept on Running

By T. Peacock

If a stationary engine on a stone or concrete foundation were out of level by the fractional part of an inch, most engineers would consider that leveling up would be highly essential for safe operation. The following instance shows what can be done in case of necessity, and the circumstances are probably without parallel.

The case referred to is that of a compressor used for furnishing blast to copper converters at the United Verde Copper Co.'s smelter at Jerome, Arizona. It is a cross-compound engine with duplex air cylinders, built by the Southwark Foundry & Machine Co. in 1902. The steam cylinders are 28- and 52-in. diameter by 60-in. stroke, while the air cylinders were originally 96-in. diameter.

In November, 1910, the air cylinders were changed to 60 in. At that time the engine was slightly out of level, but it was urgently needed, and there was no time to level it up. The cause of the engine being out of level was the subsidence of old mine workings under the power house, and several other large blowing engines and compressors were also more or less out of level. The Southwark engine was got ready for work in January, 1911, and was put on continuous 24-hour service.

Shortly after the engine was started the subsidence of the power house increased to an alarming extent, the concrete floor and walls of the building cracking in several places. A boiler room containing 16 Scotch marine boilers stood next to the power house and the caving area extended under the boilers, causing considerable trouble from leaking steam pipes and connections.

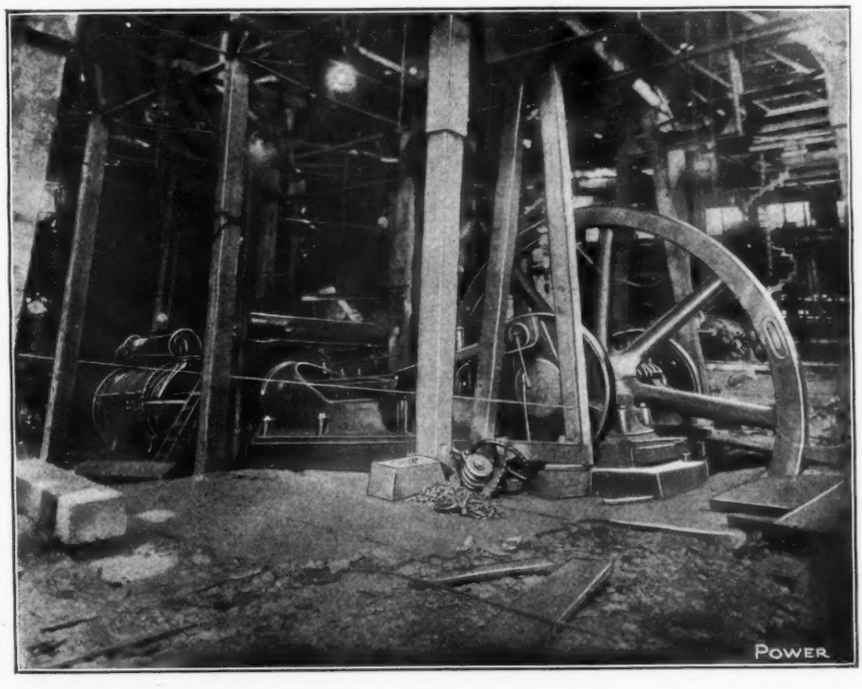
It was then decided to move the whole of the machinery and boilers from the area of subsidence to a new site as quickly as possible. New air compressors for the mine, driven by electric motors, furnished with power from a hydroelectric power plant were installed and it was only necessary to set up 10 of the Scotch boilers. These were

Result of a Return-Tubular Boiler Explosion in Mexico

moved to safe foundations, along with two Allis blowing engines for converter work. A new steam turbine-driven turbo-blower large enough to furnish blast for all the converters was ordered. Meanwhile another large Nordberg blowing engine was wrecked by the subsidence, and it was necessary to keep the Southwark blowing engine running constantly at a reduced speed, in conjunction with the Allis engines, in order to furnish sufficient blast to keep the smelter going at two-thirds of its capacity.

The Southwark engine was run steadily until March 26 of this year, when the new turbo-blower was put in operation, and it was kept in running order until April 17, when the work of moving it to a new foundation was begun. It was run for a short time as late as April, 12, and during several months it was stopped only occasionally for a few hours to lengthen the steam, exhaust, or air-discharge pipes between the point of suspension on the roof of the power house and the engine, as the subsidence under the engine was greater than under the walls of the building, necessitating swing joints in the pipes to take care of the constantly increasing distance between the main pipes and the engine, in addition to the occasional insertion of 12 or 15 in. in each pipe.

The photograph was taken on April 16, 1912, the day before the engine was dismantled for removal. The white line is a chalk line exactly level and set two feet below the original level of the floor.




ENGINE STILL RUNNING AFTER SETTLING FROM A LEVEL 2 FT. ABOVE THE CHALK LINE

The length of the engine is 41 ft. 9 in. from end to end of the bedplate, the width overall of the two bedplates being 19 ft. 6 in.

The crank end of the foundation had sunk 5 ft. 1 in. and the cylinder end was 9 ft. 9 in. below the original level, the difference in level over the length of the engine being 4 ft. 8 in. The shaft was 3 in. out of level, while the piston rods of the air cylinders were only ¾ of an inch different in level, the low-pressure steam side being down in both cases.

The engine foundation was one solid block of masonry with concrete top, the only opening in the foundation being a narrow pit for the flywheel. It is probably due to this solid form of foundation that the engine was kept running under such extraordinary conditions, as all the other engines in the same power house had separate foundations under each side of the engine, and most of them moved out of parallel during the subsidence, either by the separate foundations coming together or moving apart, a result which experience showed could not be prevented by any amount of timber struts or tie-rods.

## Large Shipment of Blowing Engines

The Mesta Machine Co., Pittsburgh, Penn., has just made a shipment of three large blast-furnace blowing engines, built for the Broken Hill Proprietary Co., Ltd., Melbourne, Australia. These engines are of the vertical high-speed type and will be used in connection with a new blast-furnace plant, which will make the first pig-iron produced in Australia.

Two of the engines are for high-pressure steam with 44-in. steam cylinders, 60-in. stroke and 84-in. air cylinders. The other engine is for low-pressure steam with an 84-in. steam cylinder, 60-in. stroke and an 84-in. air cylinder. All air heads are equipped with the Mesta plate valves, of the Iversen patent.

Twenty-three cars were required to carry this shipment, the total weight being about 1,300,000 lb. The shipment was hauled in one train to New York in 24 hr. from the time it left the yards at Pittsburgh. From New York the shipment will be taken by the American-Australian line around the Horn to Australia, the total distance being about 8000 miles and requiring about 70 days to make the journey.

**Situation for Dynamo Tender**—The United States Civil Service Commission announces an open examination for dynamo tender, on Sept. 17, 1913, to fill a vacancy in the immigration Service at Ellis Island, New York, at a salary of $900 a year. The examination will include letter writing, practical questions relating to the principles of construction and operation of dynamos and training and experience. This examination is open to all men who are citizens of the United States and 20 or more years of age. Further particulars may be obtained by addressing the United States Civil Service Commission, Washington, D. C.

**The Possibilities of Locomotive Fuel Economy** have been under investigation on the St. Louis & San Francisco R.R. for the last two years. In a paper presented before the St. Louis Railway Club recently by R. Collett, superintendent of locomotive fuel service for this railroad company, a chart was shown which indicated the results obtained by comparing the coal consumption in pounds per passenger-car mile and in pounds per thousand-gross-ton mile for corresponding months in 1910, 1911 and 1912. A material reduction was effected in 1911, and this was virtually retained in 1912 in spite of floods, severe weather, suspension of mines and other unfavorable operating conditions occurring during 1912, making it more difficult for the past year to show improvement.

# ELECTRICAL DEPARTMENT

## New Line of Portable Meters

The Westinghouse Electric & Mfg. Co. has just introduced a new line of direct-reading instruments for general testing and laboratory work where special accuracy is desired, particularly in alternating-current work. The type PC voltmeters and wattmeters operate on the moving coil principle, and the damped character of the indications permits readings to be taken quickly and accurately, which is desirable in fluctuating loads.

The movement is mounted as a unit and can be removed complete after taking off the face-plate, which makes a dust-proof joint with an inner aluminum mounting plate. The meters have a laminated iron shield riveted to the aluminum mounting plate, protecting the movement both from dust and from stray magnetic fields. The weight of the moving element is in all cases low,

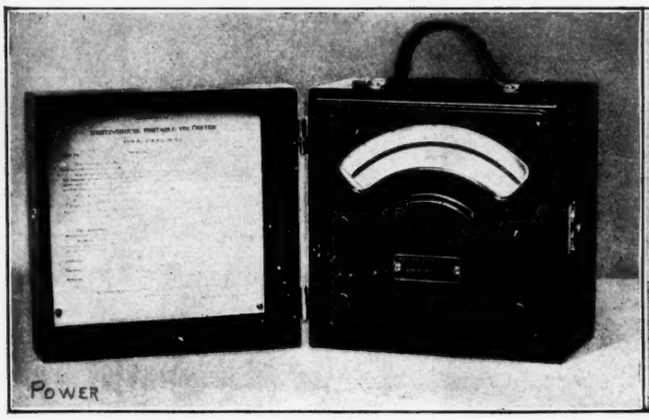



FIG. 1. INSTRUMENT WITH FACEPLATE IN PLACE

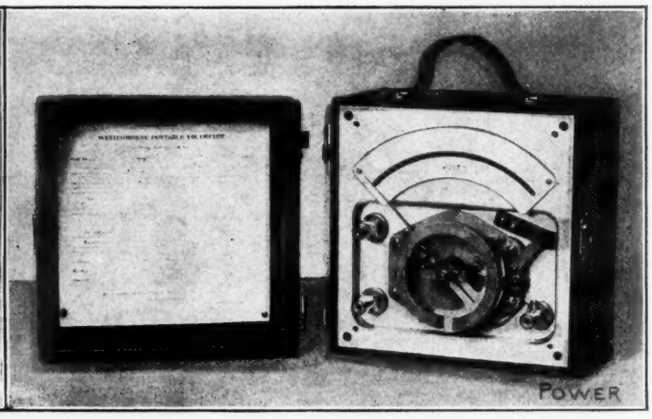



FIG. 2. INSTRUMENT WITH FACEPLATE REMOVED

preserving the pivot jewels from wear, and the torque is relatively high. The pointer is of U section, to secure strength with minimum weight; the weight of the moving element is less than 5 gr.

The meters attain their highest accuracy on alternating current, and are claimed to be free from temperature, frequency and wave-shape errors. On direct current, the meter may be subject to slight errors due to residual magnetism of the laminated iron shield, but this can be eliminated by taking the average of reversed readings.

The wattmeters are also entirely self-contained, all resistors necessary for the rated-voltage ranges being contained in the case. Those that have double current or double-voltage ranges are provided with a switch operated by a knob on the face-plate, which makes the proper series or series-parallel connection of coils when the knob is turned to the proper position. Only two current and two voltage binding-posts are, therefore, necessary

The scale for the wattmeters is wider at the lower end to give greater accuracy for the low readings, and for the voltmeters it is wider at the middle, where readings are most frequently made. A mirror extending the entire length of the scale prevents parallax in reading.

## New York Electrical Show

The Electrical Exposition and Motor Show to be opened at the Grand Central Palace in New York on Oct. 15 is to be primarily of an educational nature in relation to the use of electricity in manufacturing. The motor vehicle will be given a more prominent place than ever, and arrangements have been made to devote almost an entire floor to the electric automobile manufacturer. Besides the usual demonstration track, a fireproof model garage for the storing and maintenance of electric cars will be located on the third floor. The garage will be fully equipped with electrical vacuum cleaners, tire pumps, car-washing machines, charging boards, and current rectifiers, and will include a repair shop and battery, each fitted out with the latest electrical appliances.

Another feature of the show will be an "electric farm," occupying ninety feet or more of show space and including every kind of farm work into which electricity has been introduced. First of all will come the cow-milking exhibit. Here members of a herd of Jersey cows will be milked by an electricaliy operated milking machine and milk clarifier. The next exhibit will probably be that of an electrical dairy, where the milk will be skimmed and butter actually churned from the cream procured.

As a part of the farm exhibition a part of the space will be turned over to a landscape gardener, who will construct a miniature farm with an artificial duck pond. Here chickens and ducks will be hatched in electric incubators. Next will come an exhibition of electrical appliances used on a farm such as house and well pumps, clover cutters, corn shellers, cider mills, vegetable cutters for cattle, ensilage cutters and blowers, electrical clipping machines, vacuum-cleaning outfits and the like.

Two interesting canal displays have been arranged; one, provided by the State of New York, is a large model of the Siphon Locks on the Barge Canal. In connection with this, motion pictures of the canal in use will probably be displayed. The other canal exhibit is a 20x[illegible]-ft.

model of the Pedro Miguel Locks of the Panama Canal. The model is now in the White House.

There are a variety of Government exhibitions, including a money-making exhibit, provided by the Treasury Department; a coal-mine breaker in action, provided by the Bureau of Mines; a forestry display and an Army and Navy Signal-Service outfit. In the first-mentioned exhibition raw copper will be started through the same process used in the United States mint, the operations showing the method of rolling, cutting, milling, annealing in a furnace, washing, drying, coining and automatic weighing.

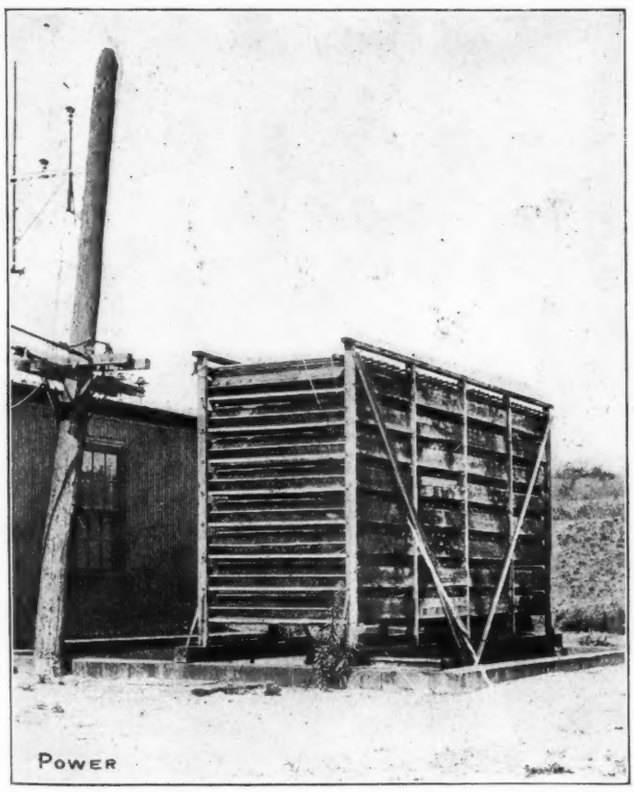
Two of the newest features of the show will be an electrical clinic, where all of the latest electro-therapeutic appliances will be on display.

Another interesting portion of the show will be the electric grill, where electricity will be used to do everything from operating the refrigerating plant to furnishing the heat in the range on which steaks and chops are grilled.

## Saving Water in a Large Substation

By H. S. Knowlton

In a substation operated by a hydro-electric power company and containing six 1500-kw. transformers reducing the line voltage from 66,000 to 13,200, the cost of cooling water for the transformers was found to be excessive.




Cooling Tower for Transformer Circulating Water

The efficiency of the modern transformer is high, running from 97 to 98 per cent. in many cases, but even this slight loss means that from 180 to 270 kw. must be dissipated in heat generated in the windings and cases of such an installation as the above. A continuous supply of cooling water circulating through pipe coils in the transformer is therefore necessary and in the operation of such substations the greatest care is taken to watch the temperature of the supply and discharge piping interiors, in order that there shall be no sudden rise in the heating effect and consequent damage to the apparatus.

In this installation the cost of city water was cut down to a small fraction of the former expense by the construction of the homemade cooling tower shown in the photograph. The hot water from the transformer coils is piped to the top of this structure and allowed to trickle down over sets of wooden racks and shelves, open to air currents and into a concrete catch basin at the bottom, from which it is forced back to the transformers by duplicate motor-driven centrifugal pumps. The evaporation has to be made up from the city mains, but a large part of the water is used over and over with substantial economy. Cooling towers are often used in steam plants, but their possibilities in electric-substation service are only beginning to be realized.

## Care of Brushes

Most instructions on the care of brushes of dynamos or motors state that one should draw a piece of sandpaper back and forth over the commutator, with the sanded side against the brush.

Just why it should be drawn back and forth, the writer could never understand, since in most brush-holders, with the exception of copper brushes, there is always a certain clearance between the brush and the holder in order to make the brush slide easily up and down and make sure contact with the commutator. Giving the matter a little thought it is easy to see that drawing the sandpaper back and forth will cause the brush to rock in the holder and thus make a loose fit between the brush and the commutator.

In the writer's opinion the sandpaper should be drawn in the direction of rotation of the commutator with a pressure on top of the brush and slightly against the rotation of the commutator. In no case should the sandpaper be drawn against the direction of rotation with pressure on the brush; on the back stroke the brush should be lifted clear of the commutator. This will make the brush fit exactly on the commutator and save considerable sparking and heating of the commutator, to say nothing of wearing grooves in it, especially when the brushes are rather hard.

Frank Garthmann.

Milwaukee, Wis.

## Electrical Power Consumed in Chicago

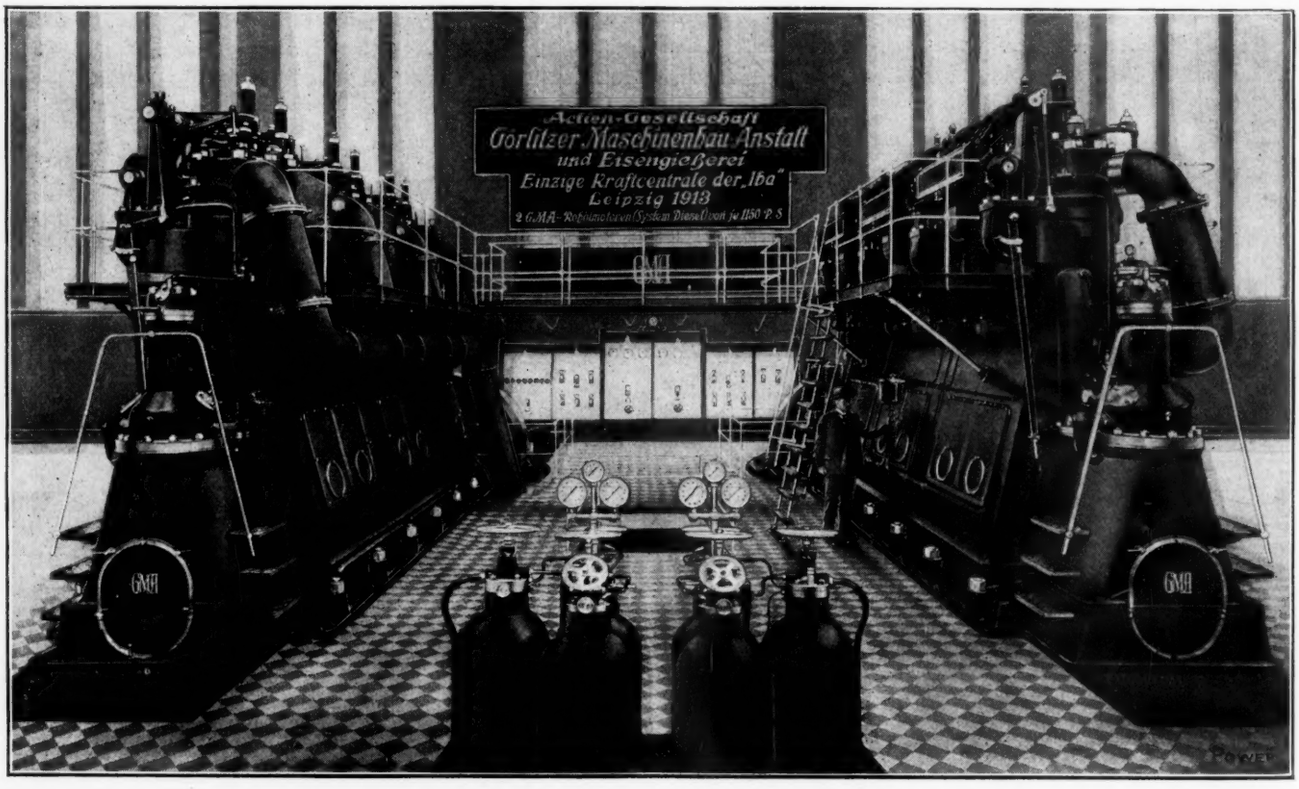
**Chicago leads the world in the production of electric power. The output of 303 electric-supply undertakings in Great Britain for the year 1911-12 amounted to 1,127,499,742 units, but Chicago by itself has an annual output of over 800,000,000, and expects within a year to turn out a billion. The Chicago Commonwealth Edison Co. generates more current than the local Edison companies of New York, Philadelphia, Brooklyn and Boston combined. The Eastern companies, however, do not supply power for local transportation, while the Chicago company does. The latter operates the largest single electric-power-generating plant in the world and its price for its product is said to be lower than that of any other electric power-vending company in the world.**

# GAS POWER DEPARTMENT

## Oil-Motor Power at Leipzig Building Exposition

By Dr. Alfred Gradenwitz

In the huge engine hall of the Leipzig International Building Exposition, two large crude-oil motors, exhibited by Messrs. Görlitzer Maschinenbau-Anstalt, Ltd., of Görlitz, aroused general interest among all visitors. These two motors are coupled directly to dynamos which supply all the electricity for light and power at the exposition.




Diesel Engines at Leipzig Exposition

The engines are of the vertical type and are arranged to work on the single-acting four-stroke-cycle Diesel principle. With a view to a satisfactory balancing of the masses, smooth running and easy starting, they are designed in six-cylinder arrangement, each engine developing 1150 b.hp. at 167 r.p.m.

A special feature of these engines is the box-shaped frame with which the cylinder jackets are cast integral. Moreover, the cylinders are cast in pairs. This design ensures a simple and compact arrangement, and the casting together of the cylinders in pairs dispenses with three bearings.

The valve gear of all the cylinders is mounted on a subdivided shaft, the bearings of which are fixed, not to the cylinder covers, but to the camshaft brackets. Therefore, in removing the valves there is no need for dismounting the valve-gear shaft, this being simply moved aside.

Each group of three fuel pumps is contained in a common casing, so that there are two pump casings for each engine, arranged jointly on one side of the engine near the governor and driven in common by a single eccentric mounted on the camshaft. Governing is effected by altering the suction-valve closure of the pumps, so as to control the fuel supply.

The single-acting double-stage vertical air pump is arranged at the side of the motor, the pump casing being fitted into the cooling water compartment of the motor frame. The pump is actuated by a crank drive from the main shaft.

The method of starting these engines is unusual, only two cylinders being used for this purpose. To this effect the lever shaft has been subdivided, only starting levers being included in the part corresponding to the two starting cylinders.

Ample water cooling of all vital parts, such as the cylinder covers, working cylinders, air-pump cylinders, etc., has been provided for. The main-shaft bearings likewise are water cooled. Lubrication of all important parts is effected from a central lubricator.

The engines are designed to operate on either mineral oils or coal-tar oils.

# Time of Ignition vs. Economy

By A. B. Morrison, Jr.

To get the maximum effects on light loads, where the compression is lower and the mixture poorer, it is necessary to have the ignition occur considerably earlier than on heavier loads. Just how much this can affect the general behavior of the engine is not generally appreciated. With the usual type of stationary engine it is not possible nor practicable to change the point of ignition to correspond to the load being carried. The latter may vary between wide ranges in such a short time that with a manually changed point of ignition the operator is unable to vary it quickly enough to correspond to the load. If, however, the ignition is automatically controlled by the governor, it can be advanced or retarded according to the load.

In order to demonstrate the difference between an engine with the ignition fixed at the same point in the stroke for all loads and one in which the ignition is automatically varied by the governor, the writer some years ago ran a series of tests, the results of which are shown in the accompanying curves. The engine was of the three-cylinder vertical type, equipped with a throttling governor and make-and-break ignition and was operated on natural gas. The time at which the igniter dropped was controlled by means of a lever connected to a collar on the governor, this lever acting on the ignition camshaft and advancing the point of ignition as the load decreased and retarding it on the heavier loads. A series of tests was run, with the governor operating in this manner, at various loads, obtained by means of a prony brake. The data secured are shown by the curves marked "I," which give the total gas per hour and the cubic feet of gas consumed per brake horsepower-hour.

A second set of readings was then taken with the same loads as in the first, but with the variable-ignition feature omitted, the igniter dropping at the same point at all loads. In making this second test the maximum load was taken at a point corresponding to 115 per cent. of the manufacturer's rating and the ignition was adjusted until an indicator diagram taken with this load showed an ignition line practically vertical. An earlier ignition would have caused the engine to knock at this maximum load. It was assumed that the engine would never be called upon to carry a load in excess of this assumed maximum. The results of the second test are shown by the curves marked "II."

An analysis of the two sets of curves shows the following:

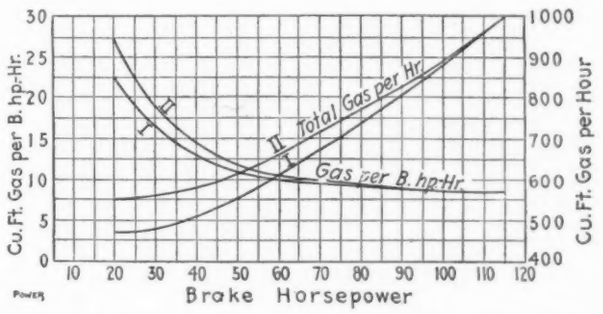
1. Between three-quarter and maximum load there is little difference in the fuel economy, whether fixed or variable ignition is used.

2. Between three-quarters and one-half load there is a considerable difference and below one-half load a very marked difference in favor of the variable ignition. Assuming the rated load of the engine to be 100 hp. (which is the manufacturer's rating), the difference at three-quarters load is approximately 4½ per cent., at one-half load, 10 per cent., and at one-quarter load, 16 per cent.

3. If the engine is designed to carry an overload in excess of the rating, and if fixed ignition is used, it should be set at a point corresponding to the maximum load which the engine will be called upon to meet. To obtain good economies on light loads, however, the ignition point at maximum load will be too early for best operating results at this maximum.

It is apparent from the curves that where an engine is operating on a load which varies widely during short intervals and with a fuel of practically constant heat value, the point of ignition automatically controlled by the governor, shows a very marked increase in fuel economy. While not apparent from the curves, the behavior of the engine at loads below 50 per cent. of the rating was much better with the variable than with the fixed ignition; the speed regulation was closer and there was no tendency to backfire, whereas, with fixed ignition, the engine was inclined to race and there were occasional backfires, though these were not pronounced.

The foregoing facts regarding fuel economy hold only with a gas of practically constant composition, such as natural or illuminating gas. Where producer gas is used, and especially with a suction producer, the variable ignition feature should not be used as it is usually a detriment instead of a benefit. This is because with a producer the gas is not likely to be even approximately uniform in quality and on light loads is usually leaner than on heavier loads. In order to obtain the same horsepower from the engine with the poorer gas more must be taken into the cylinders and the ignition must be made earlier, as the flame propagation with the poorer gas is decidedly slower. Therefore, if the engine is equipped with variable ignition and the load is suddenly increased, the governor admits more of the mixture and in so doing automatically retards the ignition. If the gas happens to be poorer than normal, which is almost invariably the case if the producer has been on light load for some time, the action of the governor in retarding the ignition is exactly opposite to what should occur, for as shown above, the poorer gas requires advancing the ignition point instead of retarding it. The result is that the maximum power is not developed from the charge and more mixture is admitted, still further retarding the ignition. The net result is an actual loss in economy and power instead of a gain. With fixed ignition this trouble does not occur to such an extent. Variable ignition controlled by the governor has been used on producer engines, but has been, in almost every case, changed to fixed ignition. The igniter is set to give the best results on about the average load, with due regard to the maximum load which the engine may be called upon to handle.




Comparison of Fuel Consumption with Fixed and Variable Ignition

# REFRIGERATION DEPARTMENT

## Mollier Diagram for Ammonia*

To facilitate the solution of refrigeration problems, a heat content-entropy diagram was prepared by the authors of the bulletin, and is presented herewith. This diagram has two families of curves: (a) curves of constant pressure, and (b) curves of constant quality in the saturated region and constant temperature in the region of superheat. The ordinates are heat contents; the abscissas are entropies.

To use as large scales as possible and at the same time to bring the diagram into reasonable compass oblique coordinates have been used. The axis of ordinates instead of being vertical is inclined to the left at an angle of 30 deg. with the axis of abscissas. Taking the values of entropy along the horizontal axis, values of heat content may be taken either along the inclined axis or along a vertical axis. Since vertical distances are equal to distances along the oblique axis multiplied by the sine of 30 deg., the scale for measuring heat contents along the oblique lines is such that a distance measured along the 30-deg. lines represents sine 30 deg. (or ½) times as many units of heat content as are represented by the same distance measured vertically.

The 30 deg. lines are lines of constant entropy and show the change of condition of the ammonia during adiabatic compression or expansion. Measurements along these lines give the change in heat content during adiabatic compression or expansion. Horizontal lines are lines of constant heat content; they show the change in condition of the ammonia which results from a throttling process, such as the passage through the expansion valve of a refrigerating machine.

The lines of constant pressure after meeting the liquid line should be extended beyond this line to the left to represent the cooling of the liquid at constant pressure below the temperature of saturation for that pressure. The constant pressure lines are so nearly tangent to the liquid line that the several lines could not be distinguished with the scale employed. The process of cooling the liquid below saturation temperature is therefore represented for all pressures by the liquid line itself.

The following examples will illustrate the method of solution of some of the more commonly occurring problems.

### Work Done in the Compressor Assuming the Rankine Ideal Cycle

In the Rankine cycle, the ammonia is admitted at a constant pressure, is compressed adiabatically to the discharge pressure, and is discharged against this constant pressure. The compressor is supposed to have no clearance. The cycle is ideal with no internal friction, no heat losses and no free or imperfectly resisted expansion. In such a cycle the work done per pound of ammonia is the difference in heat content at the beginning and end of compression. As the only compression that takes place is adiabatic, the ammonia leaving has the same entropy as that entering.

To find the work done upon one pound of ammonia, in a Rankine cycle admitting ammonia at a known quality and pressure, and compressing it to a known discharge pressure, locate the point representing the quality and pressure of the suction ammonia, and measure the distance upward to the left along a 30-deg. line to the discharge pressure line. If the quality and pressure at discharge and the suction pressure only are known, locate the point representing the quality and pressure of the discharge ammonia, and measure the distance downward to the right along a 30-deg. line to the suction pressure line. In either case the distance represents the work done in B.t.u.

*Example 1.* In a dry compression system, ammonia, initially dry and saturated at a pressure of 25 lb. per sq.in. abs., goes through a Rankine cycle in which it is compressed to a pressure of 200 lb. per sq.in. abs. How much work is done on each pound of ammonia?

The heat content of dry and saturated ammonia at a pressure of 25 lb. is seen from the diagram to be 536 B.t.u. Ammonia of the same entropy at 200 lb. pressure has a temperature of 271 deg. F. and has a heat content of 671 B.t.u. The work of the Rankine cycle is

$$536 - 671 = -135 \ B.t.u.,$$

the minus sign signifying work done on the ammonia. The same value is obtained with twice as great accuracy by measuring the distance along the 30-deg. line between the points representing the initial and final states of the ammonia.

*Example 2.* In a wet compression system, ammonia, after having gone through a Rankine cycle with a suction pressure of 25 lb., is discharged dry and saturated at a pressure of 200 lb. How much work is done on each pound of ammonia?

The heat content of dry and saturated ammonia at a pressure of 200 lb. is seen to be 560 B.t.u. Ammonia of the same entropy at 25 lb. pressure has a quality of 0.86 and has a heat content of 457 B.t.u. The work of the Rankine cycle is

$$457 - 560 = -103 \ B.t.u.$$

### Heat Rejected by Ammonia in Condenser and Cooler

To find the heat rejected per pound of ammonia in the condenser and cooler, locate the point representing the state of the ammonia after compression, and pass down a constant pressure curve to the point representing the state of the ammonia before the expansion valve. The vertical distance passed through, that is the difference between the heat contents at the initial and final states, gives the heat rejected to the cooling water per pound of ammonia. Part of this heat is taken out as the superheat, if any, is removed; the greater part is

*From Bulletin 66 of the University of Illinois Engineering Experiment Station. Authors, G. A. Goodenough and W. E. Mosher.

taken out as the ammonia is condensed; and the remainder is taken out as the liquid is cooled from the temperature of saturation at the given pressure to the temperature before the expansion valve; this latter part usually takes place in a cooler separate from the condenser.

*Example 3.* Ammonia, after being compressed as in example 1 to a pressure of 200 lb. and a temperature of 271 deg. F., is passed through the cooler and condenser and when it reaches the expansion valve has a temperature of 80 deg. F. How much heat is rejected to the cooling water?

The heat content after compression was found in Example 1 to be 671 B.t.u. Passing down the constant pressure line the superheat is removed, the ammonia condensed, and the liquid cooled from saturation temperature of 96 to 80 deg. F. At the latter point the heat content is seen from the diagram to be 54 B.t.u. The heat rejected to the cooling water is

$$671 - 54 = 617 \ B.t.u.$$

*Example 4.* Find the heat rejected to the cooling water if the ammonia compressed as in Example 2 to a dry and saturated state at 200 lb. pressure is cooled to a temperature of 80 deg. F. before the expansion valve.

By a process similar to that given in Example 3, the heat rejected to the cooling water is

$$560 - 54 = 506 \ B.t.u.$$

### Throttling through the Expansion Valve

If vapor is allowed to expand through a small orifice, such as the expansion valve, without the addition or abstraction of heat, and is finally brought to its initial velocity, its heat content will be unchanged. The proof of this statement is based on the thermodynamic theory of flow of liquids and may be found in any standard text. Horizontal lines on the diagram are lines of constant heat content and consequently show the changes in the condition of the ammonia which result from throttling in the expansion valve.

*Example 5.* Liquid ammonia in front of the expansion valve has a temperature of 80 deg. F., and after passing through the valve the pressure of the mixture of vapor and liquid is 25 lb. What is the quality of the mixture?

The horizontal line which intersects the liquid line at a temperature of 80 deg. F., intersects the 25 lb. pressure line at a point which is found by interpolation to correspond to a quality of 0.16. This horizontal line represents a constant heat content of 54 B.t.u.

### Refrigerating Effect

The refrigerating effect is measured directly by the change in $i$ (the heat content at constant pressure) which the ammonia undergoes during evaporation, this being a constant pressure process in which, as shown above, the increase in $i$ is equal to the heat absorbed.

To find the refrigerating effect per pound of ammonia, locate the point representing the state of the ammonia after its passage through the expansion valve, and pass up along a constant pressure line to the point representing the state of the ammonia at the beginning of compression. The vertical distance passed through, that is the difference between the heat contents at the initial and final states, gives the refrigerating effect per pound of ammonia.

*Example 6.* After being throttled through the expansion valve to a pressure of 25 lb. and a quality of 0.16, the ammonia evaporates in the refrigerating coils until it is just dry and saturated at this pressure. What is the refrigerating effect per pound of ammonia circulated?

From the diagram the change in $i$, or the heat absorbed by the ammonia is

$$536 - 54 = 482 \ B.t.u.$$

*Example 7.* Find the refrigerating effect per pound of ammonia when the condition after the expansion valve is the same as in Example 6, but evaporation takes place as in the wet compression system of Example 2 only until a quality of 0.86 is reached.

From the chart the refrigerating effect is

$$457 - 54 = 403 \ B.t.u.$$

### Thoretical Coefficient of Performance and Refrigerating Effect per Horsepower-hour

By definition the coefficient of performance is equal to the ratio of the heat absorbed by the refrigerating medium to the work done upon it. It has been shown that both of these quantities can be read from the diagram, and therefore the coefficient of performance can be calculated for any assumed conditions of operation.

*Example 8.* What is the coefficient of performance of the dry compression process of Examples 1, 3, 5 and 6?

From Example 6 the heat absorbed is 482 B.t.u., and from Example 1 the work done on the ammonia is 135 B.t.u. The coefficient of performance is therefore

$$482 \div 135 = 3.57.$$

Since one horsepower-hour is equal to 2546 B.t.u., the refrigerating effect per horsepower-hour is

$$2546 \times 3.57 = 9090 \ B.t.u.$$

*Example 9.* What is the coefficient of performance of the wet compression system of examples 2, 4, 5 and 7?

From Example 7 the heat absorbed is 403 B.t.u., and from Example 2 the work done on the ammonia is 103 B.t.u. The coefficient of performance is therefore

$$403 \div 103 = 3.91.$$

The refrigerating effect per horsepower-hour is

$$2546 \times 3.91 = 9960 \ B.t.u.$$

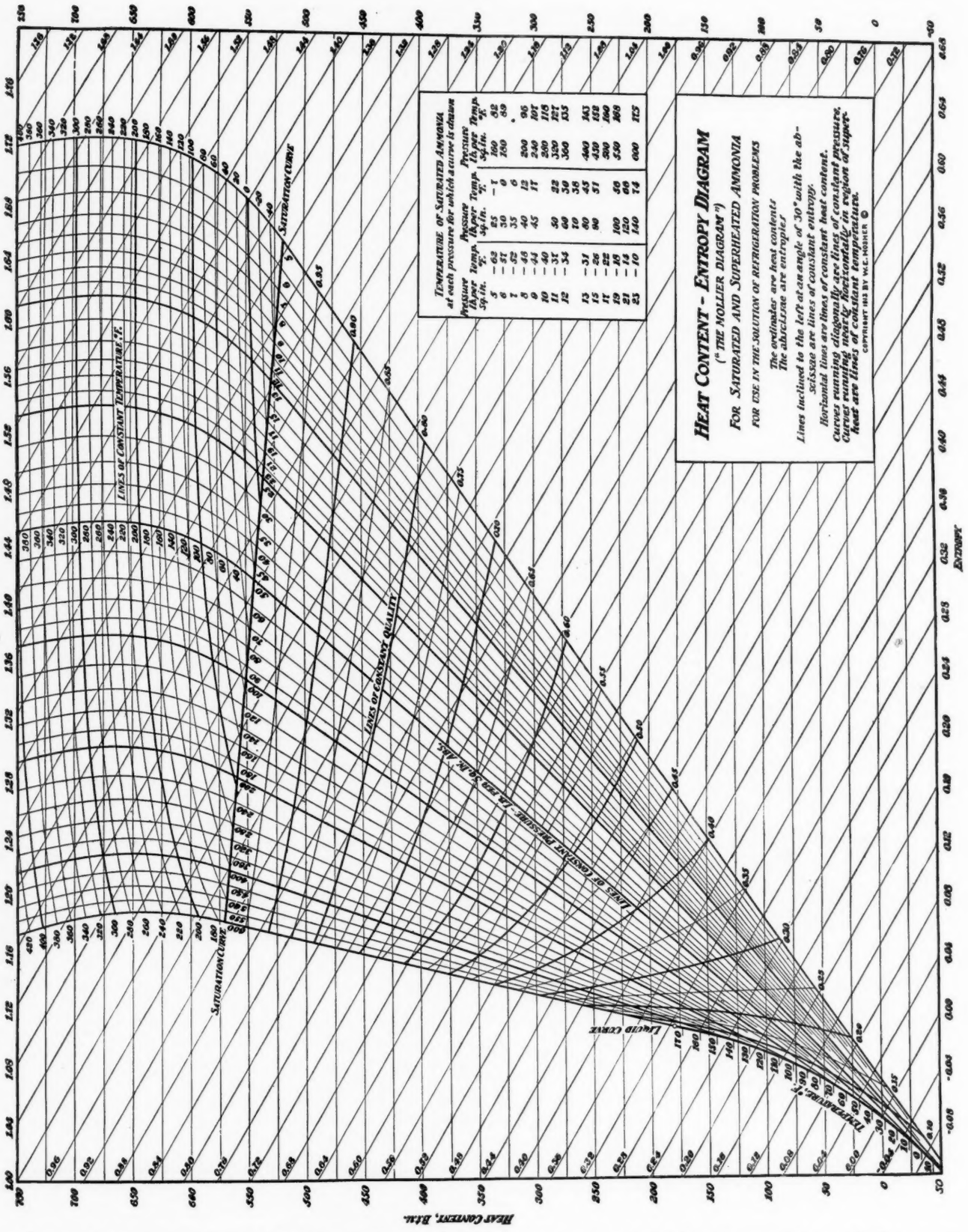
From the above examples it is seen that the theoretical loss with dry compression as compared with wet compression under these conditions is

$$\frac{9960 - 9090}{9960} = 8.7 \ per \ cent.$$

### Saving Possible by Use of Expansion Cylinder

If instead of the expansion valve an expansion cylinder were used the irreversible throttling process would be replaced by a reversible adiabatic expansion. In that case the reduction of pressure instead of being represented on the diagram by a horizontal line extending from the point representing the state before the expansion valve to the refrigerator pressure line, would be represented by an adiabatic. The length of this adiabatic would measure in heat units the work recovered by the use of the expansion cylinder and the gain in refrigerating effect would be measured by the change in $i$ along the adiabatic.

*Example 10.* Find the per cent. gain due to substituting an expansion cylinder for the expansion valve in the wet compression system described in Examples 2, 4, 5 and 7.


HEAT CONTENT - ENTROPY DIAGRAM
("THE MOLLIER DIAGRAM")
FOR SATURATED AND SUPERHEATED AMMONIA
FOR USE IN THE SOLUTION OF REFRIGERATION PROBLEMS
The ordinates are heat contents
The abscissae are entropies
Lines inclined to the left at an angle of 30° with the abscissae are lines of constant entropy.
Horizontal lines are lines of constant heat content.
Curves running diagonally are lines of constant pressure.
Curves running nearly horizontally in region of superheat are lines of constant temperature.
COPYRIGHT 1913 BY W.E. HOSMER
TEMPERATURE OF SATURATED AMMONIA at each pressure for which a curve is drawn
SATURATION CURVE
LINES OF CONSTANT TEMPERATURE, °F.
LINES OF CONSTANT QUALITY
LINES OF CONSTANT PRESSURE, LBS. PER SQ. IN. ABS.
LIQUID CURVE
TEMPERATURE, °F.
HEAT CONTENT, B.t.u.
ENTROPY

The work of compression would be the same as in Example 2, or 103 B.t.u. The heat rejected to the cooling water would be the same as in Example 4, or 506 *B.t.u.* The expansion in the expansion cylinder would be represented by a 30-deg. adiabatic extending from the point representing liquid at a temperature of 80 deg. F. downward to the 25 lb. pressure line. The length of this line gives the work obtained in the expansion cylinder, assuming a Rankine cycle, or 9 B.t.u. The refrigerating effect would be increased by 54 — 45 = 9 *B.t.u.* The coefficient of performance would therefore be

$$(403 + 9) \div (103 - 9) = 4.38.$$

The per cent gain from the expansion cylinder is

$$(4.38 - 3.91) \div 3.91 = 12 \textit{ per cent.}$$

# EDITORIALS

## Co-operation between Isolated Plants and the Central Station

Percival R. Moses, of New York, suggested in the *Engineering Magazine* last winter that the isolated plant might help out the central station at times of overload. An industrial plant need not shut down as soon as its employees stop working, but could keep on running and furnish to the distributing mains the current necessary to get them home on the tramway and to light their dinner tables, avoiding the necessity for the installation by the central station of extra plant to carry the peak which would otherwise be produced. The idea found favor in the eyes of an unknown writer, who made it the subject of an article entitled "Coöperation between Private and Public Supply Systems," in the *Electrical Review* (London) of March 28. In the August 22 issue of the same paper Douglas S. Martin points out that Moses is the apostle of the isolated plant, and takes the unknown writer severely to task for "falling to him" so easily.

Douglas S. Martin is not the apostle of the isolated plant. He would mix all the demands together so as to get the advantage of their diversity in time of demand, and then let the central station supply them all in order that it may profit by that diversity. "There is no place for the small plant," he says, "in this scheme of balance and mutual power flow, and *every small plant must go* in order to provide all the diversity for the main station which can possibly be obtained."

Diversity of demand is a good thing—enough of it will iron out all the protuberances in the load line. A comparatively small investment takes care of many maxima when they do not overlap. But the purchaser should share in the advantage. The standing charges urged in making the rate should be those on the plant actually required to meet the diversity of demand and not upon the plant which would be required to meet these demands if they were superposed, as is the case when the rate to each consumer is based upon his own maximum demand.

The load density may become so great in a given locality as to make it advisable to erect an auxiliary station to care for it. "When the auxiliary station is built and put into operation," says Mr. Martin, "it must be a big station in order that its efficiency may be high and 'industrial waste eliminated,' and it must be given as high a diversity factor as possible. To that end every isolated plant in that locality must go out of business."

If these isolated plants, so ruthlessly condemned, would be running at the time the central station is on its peak, it would not tend to flatten its peak to take them over. If they would not be running on their own load at that time, why could they not profitably be used to pump current into the mains and relieve the central station of a part of its embarrassing overload? They might not like to go out of business just to furnish diversity of demand to a central station, when they are producing light, heat and power cheaper than they could do by burning coal for the heating and buying the current.

Mr. Martin thinks that there is little hope for Mr. Moses' scheme, and predicts, basing his view on the extraordinary progress which has been made in America in the last three years in tying in main stations and systems at extra high voltages situated many hundreds of miles apart, that "in fifty years from now the isolated plant will be an extraordinary thing."

The isolated plant can have no part in this tying-in scheme, but since it is in our midst it must be used to the best advantage, and this use, to Mr. Martin's mind, would be to keep it quiet and buy current from the central station, *except on the peak,* when it should be run for the purpose of keeping down the customer's maximum demand. A man who has a plant installed and under steam ready to run, who has therefore incurred all the standing charges, which are a very considerable part of the total cost of current production, will not let it be idle the greater part of the time, unless the central station can sell him current for less than the bare price of extra fuel and labor, which is hardly believable.

⁂

## The License Situation in Jersey

After much expenditure of time and money and patience, covering a period of thirty years, the engineers of New Jersey early this summer succeeded in having a license law passed. The act became a law in the closing hours of a special session of the legislature, and this was the reason advanced for the lack of an appropriation which alone could have and can now make the law worth more than the parchment on which it is written.

Unfortunately, the bill itself made no provision for an appropriation. It simply stated that the Commissioner of Labor should, within 60 days after the act became law, organize a bureau to carry on the work of examining and licensing engineers and looking after the enforcement of the law. The act became law on July 4. The bureau should have been established on Sept. 4. But the state has done nothing more toward making the license law a reality than if it never were written. The Commissioner of Labor has done all he can without funds.

Did the framers of this law, who were so profuse with their assurances to the engineers' committee, deliberately leave out of the bill provision for an appropriation—a vital requisite? Did the able legislators intend to so write the law that it would appease the engineers and at a later date, after the hand shaking and congratulations, prove worthless? Did the Solomons at Trenton think to make sport of this last committee as it has of so many previous ones? Thirty years' experience with the New Jersey legislature inclines one to think so.

At least, the law is a reality on paper awaiting only the

funds necessary to make it actually real in practice. The next session of the legislature is months away. There is plenty of time for everyone interested to write to the assemblyman in his district, pointing out the need for this appropriation. Let the engineers of New Jersey force the legislature to show its hand.

## Flooded Cellars in New York City

New York City suffered from a cloudburst Thursday evening, Sept. 4. Many hotel and other cellars were flooded with water to a depth of seven feet and special pumping arrangements were necessary to get the water out. This is the second time this summer that serious flooding has occurred in the district between Twenty-third Street and streets along in the hundreds.

The cause is choking of the "traps" formed by looping the sewer lines under the subway.

Most hotels, apartment houses and office buildings have pumps for the heating system, though they do not have their own power plants.

Many of the buildings equipped with pumps for heating only use motor-driven pumps in which case the pumps may be quickly started. By simply putting in a tee and a valve in both the suction and discharge lines, the flood water may at least be pumped into the street as fast as it flows into the building. A couple of lengths of pipe so used would mean the prevention of much more damage, due to these choked sewers, because the conditions will surely be repeated before the volume of the sewer pipes is increased.

## Boiler Performance Curves in Boiler Room

Recently a correspondent told how valuable a chart with curves showing the kilowatt-hour output and cost of coal per kilowatt has proved in his plant when posted in the boiler room. The curves in this plant are plotted weekly and the men took such an active interest in the chart that they reduced the coal consumption materially at practically no expense to the management. Such a plan is not new, but it is altogether too uncommon, considering its potentialities for good.

It does not require argument to convince one who recognizes the peculiar psychic traits of men that the efficacy of such methods of getting men to do their best is well worth while. Every honest man who loves the approbation of his fellows wants to excel in work, in sports, in everything that he does, when he knows he has a competitor. Any fireman can acquire the skill to get as much out of boilers and fuels as any other fireman can. Unlike many of the sports, there is a limit of excellence in firing boilers that is within the reach of any man who will honestly try. Consequently, one should not fear a labor disturbance by introducing a performance chart in the boiler room. It is the shirks only who would object.

It is better to have the curves plotted every day instead of every week as our contributor has done. The interest of the men is kept more keen. It is sometimes difficult to arrive at reasonably accurate figures regarding the cost of fuel per unit of output and the enthusiasm of the men is not one whit greater than when the coal consumption per day is plotted against the output per day.

The size sheet which is most convenient for plotting these curves is fifteen by ten inches. There are sixty quarter-inch spaces on the length of the sheet and forty on the width so that by making the ordinates represent days and abscissas output, a conveniently large chart is procured. On these sheets, which may be obtained in almost any store handling such goods, there is margin enough to print the necessary letters and figures; binders may also be purchased that these sheets will just fit. If the output variation from day to day is not too great and the required data are kept, a curve representing the cost of coal per thousand or any given quantity of kilowatts, might be plotted on the same sheet after it has been removed from the boiler room.

## Encouraging Responsibility in Assistants

Ask the average engineer why he can't take a vacation, and he will reply: "Who'll keep the plant running while I am away?" Suggest that he must have an assistant, and you will probably get the answer, "Oh, yes, I've got a young fellow helping me, but he only takes care of the oiling and wiping, and does other odd jobs."

There is the reason why most engineers find things in chaotic condition upon return from an enforced absence and why they are fearsome of similar consequences when they would like to take a little time off for necessary recreation.

Every engineer should see to it that there is someone around the plant who can carry on his work at any time—one who not only can keep the wheels moving, but who can keep the plant running at its best efficiency while the engineer is absent. The logical man to be trained to this work is the man who has seen how the engineer does things—the man to whom he has explained why these things were done—his assistant.

Every chief should make it his business to explain to this understudy the whys and wherefores, discuss with him the solution of the various problems which arise, instill in him confidence by gradually assigning to him the performance of the minor duties, encourage him to carefully read and study the technical books and papers, and when the occasion arises it will be found that he is a real assistant, not one in name only, but in ability to accomplish things also.

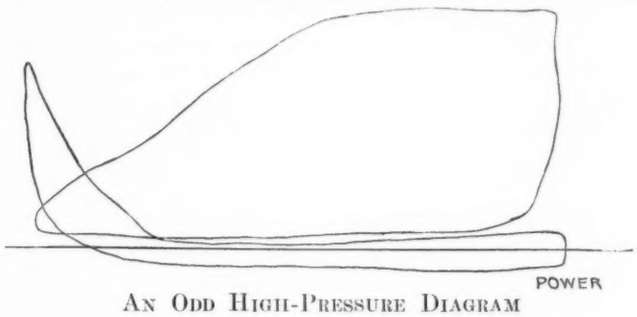
Not only will the educating of this young fellow be the source of a lot of genuine pleasure to his instructor, but the latter will find that the discussions and incidental references to standard works will refresh his own mind on many subjects, and that also it will enable him to go away for well earned recreation periods with the assurance that several weeks of hard work to get operating conditions back to normal will not be facing him when he returns.

The Federal Government has renewed its attack upon the Anthracite Coal Trust. We hope that the issue may determine not which particular set of individuals is to be allowed to bleed the industries of the country for what they can get, but the iniquity of the appropriation by any set of men of one of the necessities of our industrial life and the dealing of it out, not at a fair profit upon their actual investment, but at the price which those who must have it can be made to pay.

# READERS WITH SOMETHING TO SAY

## Peculiar Diagram

The accompanying diagrams are from the high-pressure cylinder of a Brown tandem-compound engine, running about 100 r.p.m. with 125-lb. gage pressure of steam. The valves were properly set and the engine was running smoothly and steadily. A friend of mine took the original diagrams. He simply raised the




An Odd High-Pressure Diagram

latch on the head-end steam valve while taking the card. There was a receiver pressure of 11 lb. gage at the time. Will some reader explain what caused the increased pressure in the latter end of the expansion line, also what caused the vacuum?

W. E. Chandler.

North Adams, Mass.

## Some Temporary Jobs

While making my regular trip through the storage rooms of a refrigeration plant I noticed that a 1½-in. brine pipe, through which the brine was circulating, was leaking badly enough to damage the goods in the room. Permanent repairs could not be made in less than two hours and the necessary material was not at hand. The pipe was rusted badly, so I decided to use a bandage for stopping the leak. Strips were obtained from a tightly woven bag (malt or flour bag) and rolled up and saturated with white-lead paint or cement. This was applied freely to the inside face of the strips, while wrapping them around the pipe, pulling them tight. After one layer of the saturated bag was put on a marketed cement was applied. This was continued until the pipe had four layers over the leak.

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A valve stem was attached to a 2-in. valve on a steam line supplying a cooking kettle, the valve being near the ceiling and regulated from the floor above. When the operator went to open the valve, which was shut tight, the stem broke off where it went through the floor; not much time could be spared, for the contents of the kettle had to be cooked. A broomstick bored out at one end and hammered over the broken stem, and being provided with a handwheel on the other end, served to operate the valve until there was time to renew the stem.

We have a pump on which a gland on the steam end had to be renewed. Instead of measuring the rod at its largest diameter when ordering the gland, the worn part of the rod was measured. The rod, therefore, stuck in the gland when the pump was started. The engineer not wanting to send the gland back to the shop, filed the hole bigger to fit the rod. After the pump was working packing would not hold in the stuffing-box because the hole in the gland was large and crooked. I took a hard-rubber valve disk, made it fit snugly around the rod and in the stuffing-box, after which it was cut in half and put in the stuffing-box next to the gland. The packing then held without trouble.

William L. Keil.

Philadelphia, Penn.

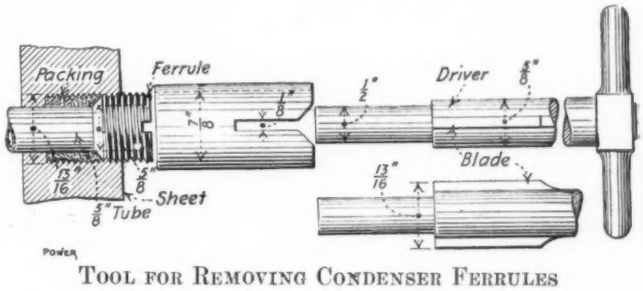
## Removing Corroded Condenser Ferrules

The condensers on a small steamer required retubing, which meant that 6000 ferrules in each condenser head had to be removed.

When attempts were made to unscrew the old ferrules it was found that they were so corroded from the action of the water that they would crumble off flush with the face of the tube-sheet.

The customary operation in a case of this kind was to break off enough of the ferrule to get rid of the shoulder formed by the $\frac{9}{16}$-in. hole in the front end of the ferrule and with a drift the same diameter as the tube, drive the tube back inside of the sheet and remove the broken ferrule with a square drift, getting enough bite with the diagonal corners of the drift to back the ferrule out.

This operation had to be repeated on each end of each




Tool for Removing Condenser Ferrules

condenser, which made it necessary to retap the 12,000 holes. The ferrule driver was T-shaped and was forged from ⅝-in. round steel. On the working end it was turned down to a loose fit in the tube for a distance of 1½ in. After this came a fin-like projection on either side of the shank projecting out about $\frac{3}{16}$ in., thus forming the blade for the slot in the ferrule.

A piece of ¾-in. pipe about 2 in. long was secured and with a hacksaw two slots about ⅛ in. wide and 1 in. long were made opposite each other, then the four

corners formed by the slots and the end of the pipes were bent in toward the center and the pipe placed on the ferrule driver. The opposite end of the pipe made a neat fit over the ferrule. The turned end of the ferrule driver was put through the old ferrule into the end of the tube and the piece of pipe brought up touching the tube-sheet inclosing the ferrule, which was then screwed out, the pipe keeping it from spreading. With this device it was possible to unscrew ferrules having but one-half of the slot remaining by striking the end of the driver a few wraps with a hammer while in place.

We removed a great many ferrules which had no slots at all, that had been broken at the starting of the job. The cost of time and material in making this tool was so trivial as to be hardly worth mentioning. The first device was made in 10 min., and was an acknowledged success, by actual count removing 60 to 70 per cent. more ferrules than by the old method and in one operation.

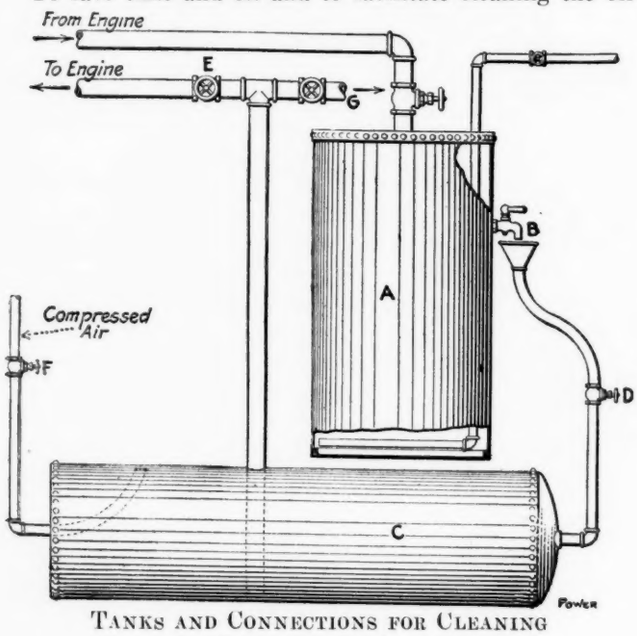
EDWARD M. DAVIS.

Philadelphia, Penn.

## Cleaning Oil from Engine Crank Cases

Our engines are all of the high-speed, inclosed type and, of course, carry all but the cylinder-lubricating oil in the crank case. As usual, it is necessary to change this oil and clean it at intervals, 10 gal. per week being taken from each crank case and boiled with steam.

To save time and oil and to facilitate cleaning the oil




TANKS AND CONNECTIONS FOR CLEANING LUBRICATING OIL

the following system was installed. Two lines of 1-in. pipe were run from the crank cases of the engines to tanks in the basement. One line was used to draw off the dirty oil from the cases and the other to return the clean oil to them.

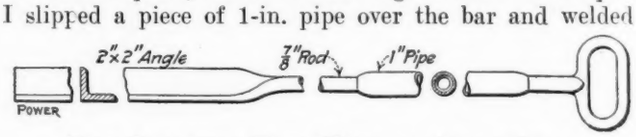
In the illustration, *A* is the boiling tank in which the dirty oil is heated. As stated above, 10 gal. of oil are run into this tank and boiled until the mixture of oil and water rises to within 2 in. of the top when the steam is shut off. As the tank *A* is lagged, it cools slowly and allows the dirt in the oil to settle. The good oil is then run from the pet-cock *B* into the tank *C*, which is an ordinary, domestic hot-water tank. Valve *D* is then closed and valves *E* and *F* opened and the oil blown to the engines as required. Valve *G* is connected to an oil-storage tank. The pipe *G* is a steam pipe and the tank *A* is partly filled with water through a pipe not shown. One man can run good oil from the tank *A* to *C*, take 10 gal. of dirty oil from any engine into the tank *A*, blow the good oil up into the engine and have the hose connections to the oil pipes cleared away within half an hour.

SAMUEL BURMFIRTH.

Montreal, Can.

## Fire-Cleaning Tool

To help the firemen in cleaning fires, I made a bar 30 in. long and of 2-in. angle iron welded to a 7/8-in. steel handle of a length to suit our fireboxes, which are 8 ft. 8 in. square, with an effective grate area of 51 sq.ft. I slipped a piece of 1-in. pipe over the bar and welded




FIRE-CLEANING TOOL MADE OF ANGLE IRON

it at both ends so that it was stiff and strong just where it passed through the firedoor. With this bar one fireman can "wing" his fires over and have the whole fire cleaned and under way in from seven to ten minutes. Two men have cleaned one fire in three and a half minutes, with from eleven to thirteen hundred pounds of dry ash on the grates.

Montreal, Can. SAMUEL BURMFIRTH.

## Cause of Knock in Pump

A more or less mysterious pound or knock will sometimes develop in pumps that have their sources of supply at a higher elevation than the suction valves. This knock is in the nature of a short jerk or kick at the beginning of each stroke. To one not familiar with such trouble, this would seem to indicate a loose piston or plunger. The cause of the knock is that the springs of the suction valves are too weak and the pressure of the water keeps the valves open until the pressure from the plunger closes them. The remedy is obvious.

Louisville, Ky. JOHN F. HURST.

## Break in Cast-Iron Suction Line

Recently a serious break occurred in the 14-in. suction pipe of our condenser. As the plant had to be kept in continuous service, I repaired the break as follows:

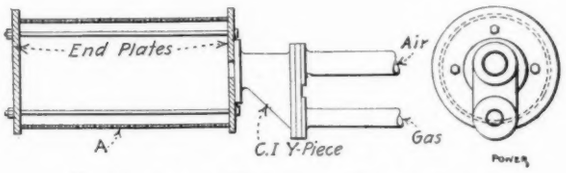
I used one quart of asphaltum thinned out with gasoline. To thicken it, I mixed in some ground asbestos until I had a thick paste. Then with a rag I cleaned the break nicely. After this was done I applied the paste until a ring about one inch thick was formed around the break. Next I bound this in with two rings of 3/4-in. tape and shortly the whole covering began to get hard. I then mixed up some ground asbestos and water and put a ring about 2 in. thick on top of this tape. For four days the condenser maintained 28 in. of vacuum with this joint in service.

Gary, Penn. W. D. LUCAS.

## Burner for Heating Crank Disks

It is not unusual to experience much trouble in heating the crank disks of large engines, preparatory to shrinking the disk on the pin. A burner for this work which gives excellent results is shown herewith.

Referring to the sketch, *A*, is a perforated plate to




BURNER FOR HEATING CRANK DISKS

the ends of which two end plates are bolted, one end-plate is fitted with a cast-iron Y-piece for supplying air and gas. The diameter of the perforated plate should be about 2 to 3 in. less than that of the diameter of the crank bore. This burner may be made in various sizes to suit the different sizes of cranks.

JOHN H. G. MORRISON.

Glasgow, Scotland.

## Abandoning a Boiler in Case of Fire

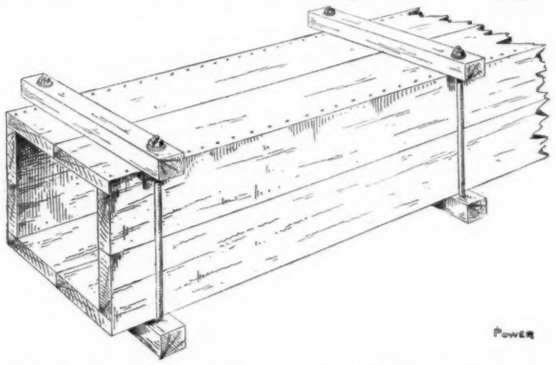
Here is a question upon which I would like to learn the opinion of other operating engineers. "What would be the best thing to do if a fire should break out in a plant using steam power, the boiler having a full head of steam on and a heavy fire in the furnace, and the fire in the building making such rapid headway that there would be no time to draw the fire?"

FRED W. HENER.

Baltimore, Md.

## Temporary Wooden Flume Cheaper than Iron Pipe

A New England manufacturing company decided to build a new penstock and dam for the water end of their power plant, and as the water for the boiler feed, condensers and industrial purposes all came through the waterwheel flume it was necessary to run a pipe line




SHOWING CONSTRUCTION OF WOODEN FLUME

from the plant to above the dam, some 800 ft. Not having permanent use for piping of the size required it was decided to build a flume of plank with the thought that the lumber would be of use after the new penstock was put in operation.

Two-inch planks, 18 ft. long, were used, costing (delivered) $30 per thousand feet. The clamps which held the flume together were 3x4-in. stock, and made of cheap hemlock lumber. The flume was 18 in. high and 16 in. wide. The wooden clamps were put on as shown, and spaced 4½ ft. apart; the rods for the clamps were of ⅝-in. common iron. The discharge end of the flume was connected to the main supply pipe in the basement of the engine room by cutting a square hole in the main pipe to receive the water from the flume.

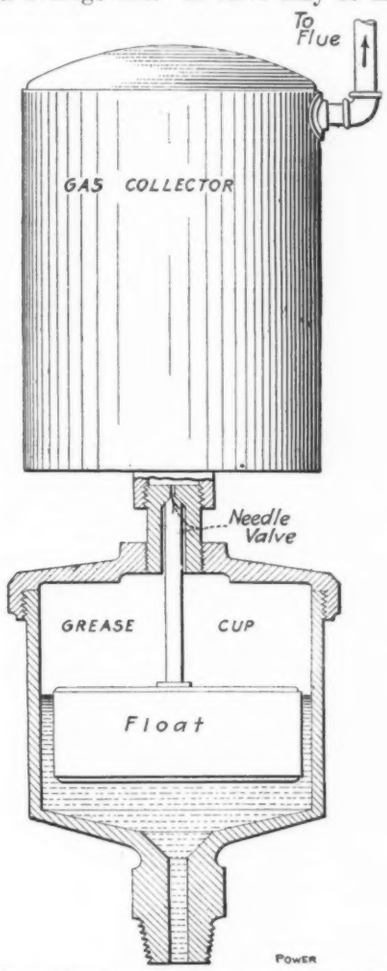
The first cost of the flume was less than the estimate of a contractor who bid for the job of putting in a cast-iron pipe 18 in. in diameter. After the flume was taken apart the planks were sold in the company's yard for $15 per thousand feet, or half the price paid for them.

L. JOHNSON.

Exeter, N. H.

## Uniform Discharge Valve

If the water in a tank or flue-gas collector is permitted to drip through a constant opening the rate of discharge varies with the head in the tank and the gas sample will not be an average one. A valve may be made from




VALVE TO GIVE UNIFORM FLOW FROM GAS COLLECTOR

an old grease cup, and connected to the tank to give a constant discharge rate with a varying head of water. The outflow is dependent upon the water level in the cup, and independent of the water in the tank.

Middletown, N. Y. R. A. BROWNELL.

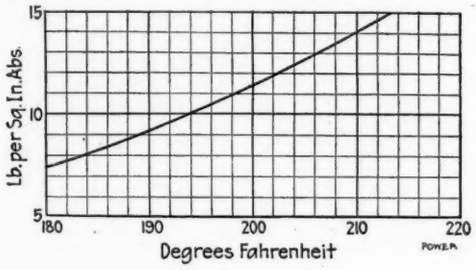
# QUESTIONS BEFORE THE HOUSE

## Measuring Water in Boiler Test

On page 830 of the June 10 number, James A. McHollan describes an arrangement for weighing feed water during a boiler test in a plant having an open heater. I wish to offer some comment upon this scheme, on the two points of feasibility and capacity.

His sketch shows the lower or suction tanks below the level of the pumps. As has been remarked by William Taylor, July 22, page 141, this arrangement will not work with water heated to nearly 212 deg. But need the water be run that hot? And how low must its temperature be held to insure proper suction?

An open heater of ample size is capable of raising water quite or very nearly to 212 deg., F., and must therefore be set high enough to supply water to the pump by gravity alone, without help from the atmos-




VAPOR TENSION OF WATER PLOTTED AGAINST TEMPERATURE

phere, because the hot water can give off vapor at atmospheric pressure. At the very least, the head from nominal water level in the heater to the pump cylinders ought to be 6 or 7 ft., and a little more is desirable if the heater may be expected to run very hot. This head is, roughly, equivalent to a pressure of about 3 lb. per sq.in.

Now the vapor tension of water, or the absolute pressure of its steam, varies with the temperature as shown by the accompanying curve, which is plotted directly from the steam table. At 200 deg., the steam pressure is about 3 lb. less than normal atmospheric, at 190 deg. it is nearly 5.5 lb. less.

Personally, I do not like to run water at more than about 190 deg. through such an open-weighing system. Hotter water gives off so much vapor as to be very annoying to the man operating the tanks and not any too good for the "works" of the scales; besides, it thus loses some weight, although perhaps not enough to be of much practical significance. With this limit imposed by outside considerations, there will be plenty of margin between atmospheric pressure and vapor tension to take care of a couple of feet of suction lift.

A question arises as to the control of this water temperature, and perhaps its temporary lowering during the test. In a condensing plant, with dependence upon returns and the exhaust from auxiliaries for heat, there is not likely to be trouble with too hot water. If the heater is of the shunt or cutout type, partial closing of the cutout valve may be effective—although this is only a tentative suggestion, not backed by experience. If all the exhaust from the main engine goes through the heater, the water must be taken as hot as it comes. Then a special cold-water pipe, fixed up so as to swing over the weighing barrels and discharging a small stream into the barrel being filled, seems a likely expedient for tempering the hot water.

After all it is a very good thing to set the heater sufficiently high at the start. The extra cost of an 8-ft. support over one 4 ft. high will be well made up when the time comes to make a boiler test.

As to capacity, the barrels shown by Mr. McHollan are presumably of the common oil-barrel size, holding 400 lb. of water. With high feed temperature, 1000 boiler horsepower will mean at least 32,000 lb. of water per hr., or 80 barrels. Then each of the two barrels will have to be filled, emptied and twice weighed in one minute and a half. Some hustling that! In fact, it could not be done with the sluggish flow due to the small heads involved.

With two barrels draining from an open heater through 2-in. valves, I should not care to undertake to handle over 500 boiler horsepower. For 1000 hp., I should put in 3-in. valves and would want tanks of at least 800-lb. capacity each, and a 1200-lb. suction tank. This would keep one man busy enough but would leave some margin for a peak load.

ROBERT C. H. HECK.

New Brunswick, N. J.

## Orsat vs. $CO_2$ Recorder

In reference to the recent letter by A. Pohlman, entitled "Orsat or $CO_2$ Recorder Best," I wish to make the following comments:

While it is true that the $CO_2$ recorder does not show CO, and it would be foolish to collect an average sample without ascertaining the percentage of CO as well as $CO_2$, nevertheless, for a spasmodic sample an engineer after a number of tests can tell whether any serious amount of CO is contained in the sample by the percentage of $CO_2$, provided that the furnace is normal, and is worked under normal conditions. An abnormal state of affairs may produce anything, as a $CO_2$ recorder gives a constant reading and not an average sample. One can keep 14 per cent. or less $CO_2$ and be safe as far as CO is concerned.

Almost everything above 14 per cent. will show some CO, and what difference does it make how much is there, the fact that it is indicates not enough excess air, and therefore the $CO_2$ reading should be brought down until no serious amount of CO is contained in the sample, which under normal furnace conditions the writer finds to be 14 per cent. $CO_2$ or 50 per cent. excess air.

It is true that it is impossible to cheat the collector,

so long as the machine collects a uniform sample of gas. The ordinary bottle turned upside down, allowing water to drip out, thus taking in the gas, is easy to cheat, as three or four times as much water goes out in the morning as in the afternoon, due to the difference in head pressure. [See valve, page 405.—EDITOR.]

It is an easy thing to get gas into a bottle and get it out again for analysis, but it is another thing to get a correct average sample. The bottle scheme for collecting gas is unscientific and unsatisfactory. In a large Eastern plant they have a number of bottle collectors, and an actual test proved that they could not tell from the sample in the bottle within 2 per cent. $CO_2$ what the real average was. For example, if the bottle sample analyzed 10 per cent., the true average might be as low as 8 or as high as 12 per cent.

While Mr. Pohlman's statement regarding several other troubles with $CO_2$ recorders, such as weakening of the absorption solution, breakage, etc., may be true of some older types, the later types certainly do not cause trouble. The writer knows several of one make that have run for about a year without being touched, outside of changing charts each day and charging with caustic solution every other week. These machines have been checked up with the portable flue-gas analyzer and the solution has never failed to take out all the $CO_2$.

The Orsat was designed by a French chemist about 75 years ago. It is substantially the same now as it was then. There have been numerous modifications of the Orsat by chemists who never intended the instrument for engineering uses. There is no engineering supply house dealing in Orsats. The laboratory houses all sell them, and some engineers buy them because they do not know that there is an engineer's analyzer on the market—an instrument designed by an engineer and especially adapted for the use of engineers.

WALTER C. EDGE.

Collingswood, N. J.

## Steam-Tight Valves?

In connection with the report on page 157 of the July 29 issue of tests on a 13x15-in. Skinner engine showing a nearly constant steam rate from ¼ to 1¼ load, the reason being found in a steam-tight valve, it may be of interest to call attention to the first report of the Committee on Steam-Engine Research of the Institution of Mechanical Engineers, Mar. 17, 1905. The report was based on an extensive series of tests to determine, among other points, the amount of leakage past slide valves and the effects of condensation and reëvaporation. The steam ports on the engine used for the experiments were blocked and it was run by power. The valve was of the Meyer expansion type. The cylinder heads and barrel were jacketed.

It was found that the leakage from the steam chest to the exhaust was nearly proportional to the square roots of the differences in pressure, that it was less with good lubrication, less with increased overlap of the valve, and that with well fitted valves, the amount of leakage may be over 20 per cent. of the steam entering the cylinder and is rarely less than 4 per cent. *Machinery* for May, 1905, page 483, gives a digest of the report.

CHARLES H. CHASE.

Stoneham, Mass.

## Preventing Air-Driven Pumps from Freezing

On page 138 of the July 22 issue, H. E. Leonard raises the question of how to prevent an air-driven pump from freezing without heating the air.

In a recent issue, F. M. Benjamin offers two suggestions. Without wishing to seem to criticize Mr. Benjamin, the remedies proposed are inadequate to meet the situation as it confronts Mr. Leonard.

The first suggestion made of allowing the pump to stop for five minutes and thaw out would never avail except under circumstances requiring only irregular service. The second suggestion of spraying water through the exhaust pipe by means of ⅛-in. pipe connecting with the discharge of the pump, will only prevent freezing in that portion of the exhaust in which the water is kept flowing. It will not help matters back at the cylinder where water is actually freezing, and gradually closing the exhaust passages. In cold weather spraying water into the exhaust pipe would, perhaps, soon cause the outlet to be frozen entirely.

In the usual installation the supply pipe drains toward the pump and a drip valve should be placed to take care of the condensed vapor before it reaches the throttle. The supply line should be blown through prior to starting the pump. Immediately the air passes the throttle a rapid expansion takes place with resulting cooling and ice will form on the slide valve and in the passages.

A mechanical mixture of oil and water in a semi-saponified condition is difficult to freeze and even if frozen the resulting substance is of a flaky structure and very readily disintegrated.

The following method is dependent upon the above statement, I have tried it out on pumps operating in zero weather and met with success.

Place an ordinary sight-feed lubricator immediately beyond the throttle and on the pump side of it. Make up a mixture consisting of one part kerosene, one part engine oil and one part soap suds and violently shake until an unctuous, white liquid is obtained. Use this in the lubricator, allowing a feed of about three drops per minute.

Snow flakes will be formed and blown out of the exhaust pipe, but no ice will adhere to the surfaces of the valve chest or cylinder.

E. HURST.

Boonton, N. J.

## Why Engines Rock

In the Aug. 5 issue, John F. Hurst, in answer to an inquiry by H. R. Low, states that the reason for excessive rocking of an engine in most cases is caused by too much compression. Mr. Hurst by no means stated a number of other important and probable causes for such trouble. Some of these causes are late admission, no compression, unbalanced engine, poor foundation.

I was once called to correct a rocking engine which was direct-connected to an electric generator. At every revolution of this machine, it seemed as though it would jump from the foundation. This unit had a new concrete foundation, and the erecting engineer said that the trouble was due to the construction of the foundation,

while the concrete contractor blamed the engine. I applied the indicator and found that the engine did not take steam until almost one-half stroke. The trouble was caused by the governor being put together wrongly, so that all parts of the stroke were extremely late. On reassembling the governor, however, a very good diagram was obtained, and the vibration had ceased.

FREDERICK L. RAY.

Louisville, Ky.

## Steam-Expanded Piston Rings

The issue of Aug. 26 shows illustrations of Mr. Gulbranson's steam-set piston rings. There is nothing new in connection with these rings, except the number of parts in construction, viz., the double-packing ring and the carrying ring beneath.

I used steam-set rings more than 20 years ago. I do not consider them a good thing for several reasons, the most serious one being that a cylinder wears out of size at the point of greatest pressure, or the points from admission to cutoff.

I bored one cylinder that was $\frac{3}{16}$ in. larger at each end than in the center, and needless to say the piston was not drilled with small holes for steam under the rings in any of the refit jobs.

BURT M. SEYMOUR.

Delphos, Ohio.

## Corliss versus Four-Valve Engine

I read with interest C. R. McGahey's article on the above subject in the issue of July 8.

Mr. McGahey makes the following statement: "The thermal loss is greater in the four-valve engine, as the temperature of the cylinder is changed about twice as often as with the Corliss engine." This is so fundamentally incorrect that I cannot refrain from commenting upon it. The high-speed engine should theoretically have a better steam economy than the low-speed machine. One of the greatest sources of loss in the cylinder is due to the initial condensation of the entering steam. The low temperature of the exhaust steam cools the cylinder wall to a temperature considerably lower than that of the entering steam. This cooling of the wall requires a definite amount of time, and time is a very important factor in the amount of condensation and not the number of changes. With sufficient time the temperature of the wall would be reduced to that of the exhaust steam. Before such a reduction in temperature can take place the wall is again exposed to the high temperature of the entering steam. The rapidity with which this occurs, or in other words, the less time between high-temperature baths, the higher will be the wall temperature and the less will be the initial condensation. It is obvious that if the rotative speed is high the time which the cylinder wall will have for cooling off will be less, therefore there will be less cylinder condensation. Consequently cylinder condensation is theoretically less in high-speed than in slow-speed engines.

This principle has so long been recognized that I was much surprised that Mr. McGahey could have been so confused as to make the statement that the loss is greater "as the temperature of the cylinder is changed about twice as often." The author may possibly have had in mind that the same steam was always passing through the cylinder when evolving the theory that the number of changes in a given time are the determining factor. It has long been recognized as a fundamental principle in thermodynamics that if two elements are brought in contact, one having a higher temperature than the other, the one will give out heat which will raise the temperature of the cooler element and the cooler will give out "cold" (so to speak; strictly expressed, will absorb heat), and this will continue until the two are finally of the same temperature. This transfer of heat and "cold" requires time.

There are, of course, other factors than cylinder condensation that determine the final results as to steam consumption between the slow-speed and the high-speed engines, such as clearance, port opening, etc., but the high-speed machine undoubtedly makes a gain over the slow-speed in cylinder condensation. There is also another factor in favor of the high-speed machine regarding the cylinder condensation, and that is that the amount of cylinder area in the slow speed is greater than in the high speed, therefore the condensation loss will be greater, due to the greater radiating surface. In comparing the size engines mentioned in the article, 15½x18-in. four-valve at 220 r.p.m. with the 18x36-in. Corliss valve at 90 r.p.m., which would give about the same horsepower, the cylinder area of the Corliss-valve engine is about twice as great as that of the four-valve engine.

I assume that the indicator diagrams shown in the article have no relation to the tests reported as the four-valve diagram shows about 40 per cent. cutoff, which presumably is not the point of highest efficiency for the particular engine, whereas the Corliss shows about 20 per cent. cutoff, which is very close to the point of highest efficiency. From my experience and knowledge the modern high-grade four-valve (Corliss-valve non-releasing gear) engines are giving results better than the slow-speed Corliss engines. I have knowledge of a number of very complete and correct tests which have been made on four-valve engines. For example, one was a 240-hp. four-valve engine operating with 110-lb. initial steam pressure exhausting to the atmosphere, 220 r.p.m., and the steam consumptions in pounds of dry steam per indicated horsepower for different percentages of rated load were as follows:

| Percentage of load | 25 | 50 | 75 | 100 | 125 |
|---|---|---|---|---|---|
| Lb. per i.hp. | 24.7 | 21.25 | 20.9 | 21.17 | 22.9 |

It is very doubtful if results as good as these have been obtained on the slow-speed Corliss engine of the same horsepower rating operating under the same conditions.

I also have knowledge of several plants where periodical economy tests have been made on four-valve engines covering a period of years, showing that the steam consumption has not increased to any extent, in fact in some cases the steam consumption decreased after a period of considerable operation, which would tend to prove that the valves were tight.

S. F. FERGUSON.

New York City.

[Previous discussion of this subject will be found in the issues of POWER, September 24, 1912, April 8, 1913 and April 15, 1913.—EDITOR.]

# INQUIRIES OF GENERAL INTEREST

**Producer-Gas-Engine Results**—What is the best average result with a producer-gas engine in pounds of coal per brake horsepower-hour?

W. G. M.

About 1½ lb., but most plants use nearer two pounds.

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**Steam Line Draining**—Should a steam line to an engine be arranged to drain toward the engine or back to the boiler?

W. G. M.

The general rule for a steam line is to drain in the direction of flow of the steam, but it should drain from the first or boiler stop valve in both directions.

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**Operating Turbine Without Dry-Air Pump**—Can any type of turbine connected to a barometric high-level counter-current jet condenser be kept running without an air pump?

M. F. K.

In case of failure of a dry-air pump on an ordinary barometric condenser the vacuum would be reduced and in a counter-current jet condenser the action would be the same, the air being drawn down with the discharging water. The turbine could be kept running but not with the same load and steam pressure as the vacuum would be reduced.

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**Oil for Draft Gages**—What is the best nonvolatile oil for use in draft gages?

W. G. M.

Thin machinery oil makes a good nonvolatile oil for a draft gage. For an ordinary V-form of draft gage, if the gage is partly filled with water and an equal depth of oil is placed on the surface of the water in each leg of the gage, evaporation of water is prevented and the difference of level can be read off as inches pressure of water column without allowance for the specific gravity of the oil, as would be required in reading a tube containing only oil.

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**Combined Boiler and Furnace Efficiency**—How is the combined efficiency of a boiler and furnace found, when the heat value of the coal and the weight of water evaporated are known?

J. W. D.

To evaporate 1 lb. of water from and at 212 deg. F., 970.4 heat units are required. If N is the number of pounds of water evaporated from and at 212 deg. and C the number of British thermal units to be realized in perfect combustion of a pound of coal, then the combined efficiency of furnace and boiler would be

$$\frac{N \times 970.4}{C}$$

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**Lifting Hot Water**—What is the highest temperature at which water can be lifted by the suction of a feed pump to a height of 5 ft.?

F. D. S.

The temperature of vaporization of water at the absolute pressure of the pump suction. Lifting 5 ft., the pressure of the water column being

$$5 \times 0.433 = 2.165 \text{ lb. per sq.in.}$$

the pressure in the upper surface of the column would be

$$14.7 - 2.165 = 12.535 \text{ lb. per sq.in.,}$$

and the highest theoretical temperature at which water could be lifted would be 204.1 deg. F. Actually water cannot be lifted 5 ft. at this temperature because of the liberation of air from the water, occasional vaporization of the water, and leakage of air into the pump or suction pipe causing the pressure on the top of the column of water to be in excess of the theoretical amount of such pressure.

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**Injector or Inspirator**—What is the difference between an injector and an inspirator?

H. H. S.

In marine practice and in some localities the terms injector and inspirator are used interchangeably. Commercially the term inspirator is almost exclusively confined to the product of one company, which differs in some important respects from the instruments commonly classed under the head of injectors. It consists essentially of a lifting-jet and lifting nozzle or injector proper. When the inspirator is in full operation steam is admitted to the first nozzle to deliver the supply water to the force nozzle by which latter the water is forced into the boiler. Although the lifting and force nozzles are fixed their proportion to one another is such that the instrument requires no adjustment for changes in steam pressure or water supply, the waste valve being kept closed while the instrument is in operation except when starting the operation of the force nozzle. There are other types of double-barreled injectors, but only one goes under the commercial name of "inspirator."

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**Steam Consumption from Indicator Diagram**—How can the steam consumed by an engine be determined from an indicator diagram?

F. E. B.

The simplest method is to use the formula:

$$M = \frac{13{,}750}{\text{Mean effective pressure}} [(C + E) \times W_e - (H + E) \times W_h]$$

which gives the weight in pounds per indicated horsepower per hour

C = Proportion of the stroke completed at a point on the expansion line which should be taken near the point of actual cutoff or near the point of release.

H = Proportion of compression.

E = Proportion of clearance.

$W_e$ = Weight of 1 cu.ft. of steam at the pressure of the point considered.

$W_h$ = Weight of 1 cu.ft. of steam at the compression pressure.

We and Wh are to be found from tables of the properties of steam.

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**Boiler-Opening Reinforcement**—How should the relative dimensions of a reinforcement for a hole in the barrel of a boiler for a dome opening be figured?

W. N. A.

The opening being oval or elliptical in shape, not over 17 in. long and 13 in. wide, with the longer axis placed in a circumferential direction of the shell, a strengthening ring should be riveted around the hole (generally placed inside of the shell) and the cross-section of this ring should be at least as strong as the longitudinal seam of the boiler. The Massachusetts and Ohio regulations require such rings to be of wrought iron or cast steel with a net cross-sectional area, taken on a line parallel to the axis of the shell, not less than the cross-sectional area of the plate removed on the same line.

For instance, if a 17x13-in. dome opening is made on the top of a boiler with the length of the opening circumferentially of the shell, and the boiler shell is made of ½-in. material, then

$$½ \times 13 = 6½ \text{ sq.in.}$$

will be the greatest cross-sectional area of shell removed. One side of the reinforcing ring, therefore, has a net cross-sectional area of

$$6½ \div 2 = 3¼ \text{ sq.in.}$$

If the ring be ¾ in. thick and riveted with two rows of $\frac{13}{16}$ = 0.8125 in. diameter rivets, then as each rivet would remove 0.75-in. thickness of ring × 0.8125 in. diameter of rivet = 0.609 sq.in., and assuming that there are two rows of rivets which, although staggered, should both be assumed as in the same cross-section of the ring, then the gross cross-section of the ring should be

$$3.25 + (2 \times 0.609) = 4.468 \text{ sq.in.,}$$

and the ¾-in. thick ring would be

$$\frac{4.468}{0.75} = 5.95 \text{ in.,}$$

or practically 6 in. wide for the ring to restore the full strength of plate removed.

If the ring is to equal the strength of the longitudinal seam, then multiply the width (found to be 6 in. as above) by the efficiency of the longitudinal seam of the boiler.

# ENGINEERS' STUDY COURSE

## Evaporation—IV

### Solution of Practice Problems

The factor of evaporation for the conditions specified in the problem given last week may be ascertained by means of the formula, also given last week:

$$F = \frac{(H - t) + 32}{970.4}$$

By referring to the steam table given in the issue of Sept. 2 it will be seen that the total heat of steam at 120-lb., abs., pressure is 1189.6 B.t.u. Then substituting in the formula, we have

$$F = \frac{(1189.6 - 160) + 32}{970.4} = \frac{1061.6}{970.4} = 1.094$$

A boiler horsepower is equivalent to the evaporation of 34.5 lb. of water per hour from and at 212 deg. As explained last week the equivalent evaporation for any set of conditions is found by multiplying the actual evaporation by the factor of evaporation. Thus, the equivalent evaporation for the conditions given in the problem is

$$8670 \times 1.094 = 9485 \textit{ lb. per hr.},$$

and, hence, the number of boiler horsepower being developed is

$$9485 \div 34.5 = 275.$$

### Steam Calorimeters

It is important to know the quality or the percentage of dryness of steam when comparing boiler or engine tests. If this were not known and taken into consideration the results of some tests would be misleading.

Moisture in steam causes inefficiency and when present in large quantities it is troublesome in the steam engine as it makes cylinder lubrication difficult and may cause knocking. It also tends to cause water-hammer in the pipes.

The device used to discover the percentage of moisture in steam is called a steam calorimeter. It is not like the fuel calorimeter (mentioned in the issue of June 10) which is used for estimating the heat value of fuels and the two should not be confused.

There are three principal types of steam calorimeter. One is the barrel, in which the steam sample is injected into a barrel of cool water and the increase in temperature and weight noted, from which data the quality of the steam can be estimated as in the lesson of Sept. 9. This type is not popular, however, because very slight errors in reading temperatures and weights cause large errors in the final results.

Another type is the separating calorimeter. In this the particles of moisture are separated from the steam by mechanical means in a manner similar to that used in an ordinary steam separator in a pipe line.

The third type is the throttling calorimeter, a form of which is illustrated herewith. Where the percentage of moisture in the steam does not exceed 4 per cent. and where the steam pressure is 80 lb., gage, or over, the throttling calorimeter is very accurate and convenient, hence, its use is preferable. When the contained moisture is greater than 4 per cent., this form of calorimeter is useless and one of the separating type must be used.

As the quality of the steam in the great majority of plants seldom falls below 96 per cent., only the throttling type of calorimeter will be taken up here. If its principle and operation are fully understood, no difficulty whatever will be experienced in mastering the principle and operation of the other types.

When steam in expanding forces the piston ahead in the cylinder of an engine it loses heat, which is spent in doing the work of making the engine turn and running the dynamo, shafting system or whatever the load-giving apparatus happens to be. On the other hand, when steam is permitted to expand from one pressure to another without resistance, as it does when flowing from a hole in a pipe or from a valve into the open air, it tends to retain all the heat it had.

To illustrate, assume that a boiler is generating steam at a pressure of 140 lb., abs., and that this steam is dry saturated. By referring to the steam table in the study course of Sept. 2 it may be ascertained that steam of that pressure has a total heat of 1192.2 B.t.u. per lb. of steam. If this steam were allowed to flow through an opening directly into the atmosphere, we would find that the temperature of the steam after it emerged from the opening, and its pressure had dropped to that of the atmosphere, was 302.9 deg.—that is, if we were careful to avoid error due to radiation loss. The temperature of saturated steam at atmospheric pressure is 212 deg. Then, this steam must be superheated

$$302.9 - 212 = 90.9 \textit{ deg.}$$

The total heat of dry saturated steam at atmospheric pressure is 1150.4 B.t.u. per lb. This superheated steam has all the heat of dry saturated steam and some more, as its higher temperature indicates. The specific heat of superheated steam varies for different pressures and temperature, but for the above case would be about 0.46. Hence, the excess heat must be

$$90.9 \times 0.46 = 41.8 \textit{ B.t.u. per lb.}$$

And this value added to the total heat of dry saturated steam at atmospheric pressure gives

$$1150.4 + 41.8 = 1192.2 \textit{ B.t.u.},$$

which is exactly the same as the heat contained in 1 lb. of the steam we started with, namely, dry saturated at 140 lb., abs., pressure.

Now, if the 140-lb. steam contained some moisture, the total heat in the mixture of steam and water would be less to begin with, and the difference between that heat and the total heat of steam at 212 deg. would not all go to superheating the steam at atmospheric pressure, but some of it would be used up in vaporizing the entrained moisture and the final temperature or degree of superheat would, therefore, be considerably lower than above shown.

For instance, suppose the quality of the steam were 98 per cent., that is, it contained 2 per cent. of moisture. The total heat of a pound of such a mixture of steam and water would be the heat of the liquid plus the latent heat of 0.98 lb. of steam, or

$$324.6 + (867.6 \times 0.98) = 1174.8.$$

Subtracting from this, the 1150.4 B.t.u., total heat in 1 lb. of dry saturated steam at atmospheric pressure, we have

$$1174.8 - 1150.4 = 24.4 \ B.t.u.$$

excess heat instead of the 41.8 we previously had. Hence, the degree of superheat would be

$$24.4 \div 0.46 = 53 \ deg.,$$

showing a final temperature, consequently, of

$$212 + 53 = 265 \ deg.$$

Thus, by permitting steam to flow through an opening into the atmosphere, observing the temperature change, and using precaution against error due to radiation, we may calculate the quality or moisture content of such steam. A home-made form of the throttling calorimeter—the device used in doing this—is shown in the accompanying illustration. It can be made of pipe and fittings as shown. Between the flanges tightly fit a ⅛-in. steel plate with a small orifice drilled through its center. This orifice should not be greater than $\frac{3}{32}$ in. in diameter; $\frac{1}{16}$ in. would be better. The thermometer wells can be obtained from any instrument supply house. One concern quoted the writer a price of $1.60 apiece for the size specified in Fig. 1. This price includes the mercury filling for the well. The thermometers should have a range up to 400 deg., F., at the very lowest. The same concern quoted a price of $3.50 each for thermometers of this range. While the calorimeter can be operated with fair success with only the lower thermometer and well, it is preferable to use both an upper and a lower; the increased reliability of results more than offsets the difference in cost.

When assembled and in place, the calorimeter should be well insulated with magnesia or other covering to eliminate radiation as much as possible and, hence, prevent error.

According to the boiler-testing code of the American Society of Mechanical Engineers, formulated in 1899, the nozzle *A* of the calorimeter "should be placed in the vertical steam pipe rising from the boiler. * * * and should extend across the diameter of the steam pipe to within ½ in. of the opposite side * * * none of these holes (in the nozzle) should be nearer than ½ in. to the inner side of the steam pipe."

Before any readings are taken it is best to let steam blow through the calorimeter at full pressure for 5 or 10 min., so that the entire apparatus will be thoroughly warmed up and error avoided. The thermometers should be tested for accuracy before used and if an important variation is found allowance should be made in the readings. For very accurate work corrections are made for the error caused by the exposure of part of the thermometer stem to the room temperature. But, such connections are far too small to be of real importance in every-day power-plant test work. So, they will not be considered here.

When making a test for the quality of steam have the valve of the calorimeter wide open and read both thermometers. The temperature shown by the upper thermometer should correspond very closely with the temperature of steam at the pressure being carried on the boiler. This temperature is given in the steam table. If an appreciable difference is found this may be taken as an evidence that either the steam-pressure gage on the boiler or the thermometer is inaccurate and both should be tested to locate the trouble.

After the readings have been taken, the quality of the steam may be estimated by the following formula, which appears more formidable than it really is.

$$Q = \frac{(H - h) + 0.46\,(T - T_1)}{L}$$

where

$Q$ = Quality of steam expressed as a decimal fraction;
$H$ = Total heat of dry saturated steam at atmospheric pressure, B.t.u. per lb.;
$h$ = Heat of the liquid corresponding with the pressure carried on the boiler, B.t.u. per lb.;
$T$ = Temperature as shown by the lower thermometer degrees, F.;
$T_1$ = Temperature of saturated steam at atmospheric pressure, degrees, F.;
$L$ = Latent heat of steam at the pressure carried on the boiler, B.t.u. per lb.

To illustrate the use of this formula, assume that the boiler being tested is at sea level and that the temperature shown by the upper thermometer is 366 deg. and by the lower one, 266 deg.

Referring to the steam table, we would find that the steam pressure corresponding with the temperature of 366 deg. is 165 lb., abs. If this figure checks up reasonably close with the pressure as shown by the gage on the boilers, we may proceed with our calculating. In making the comparison, don't forget to subtract the atmospheric pressure from the figure showing the absolute pressure, in the present case, 14.7 lb.

All the factors, except $Q$ and $T$, are given in the steam table; $T$, given by the lower thermometer of the calorimeter, is 266; $Q$ will be the answer. Substituting,

$$Q = \frac{(1150.4 - 338.2) + 0.46\,(266 - 212)}{856.8} = \frac{837.04}{856.8}$$

$$= 0.977$$

That is, the quality of the steam is 97.7 per cent.

### Practice Problem

At a plant considerably above sea level, the atmospheric pressure is 13 lb. per sq.in. A boiler was being tested for the quality of the steam generated. The upper thermometer of the throttling calorimeter used showed a temperature 377.6 deg., while the lower thermometer showed 286 deg. The steam-pressure gage on the boiler indicated a pressure of 177 lb., gage. What was the quality of the steam?

❖

**Ventilation Problems** are to be studied exerimentally during the next four years, under the direction of the following commission: Prof. C.-E. A. Winslow, of the College of the City of New York; Prof. F. S. Lee, of the College of Physicians and Surgeons, Columbia University; Prof. E. L. Thorndike, Teachers College, Columbia University; Prof. E. B. Phelps Massachusetts Institute of Technology; Dr. James Alexander Miller, New York City, and D. D. Kimball, of Richard D. Kimball Co., 15 West 38th St., New York City. A sum of $50,000 has been appropriated for the study, from a gift made by Mrs. Elizabeth Milbank Anderson to the New York Association for Improving the Condition of the Poor.

# Soda-Ash for Boiler Waters*

By Charles H. Bromley

*SYNOPSIS—The article deals chiefly with the troubles most often experienced when using soda-ash in boiler waters. The effects of diluted solutions are pointed out and a normal solution is defined. The reason why steam mains and valves corrode while the boilers are unaffected by the soda-ash is also explained.*

The corrosion of boilers is, without doubt, the most serious problem that confronts many engineers. Among the "home" remedies none are so common and at the same time more misused than soda-ash. Before stating something of what is known about soda-ash, the following serves well as an example of one of those puzzling cases of corrosion that sometimes occurs.

### Influence of Sulphur in Boiler Plate

It is well known that sulphur is one of the most corrosive impurities in steel. The sulphur in boiler plate is distributed over the metal in minute spots which are claimed by many to form centers in which corrosion begins, and experience in a plant of the London Electric Supply Corporation seems to verify the report:

A standby boiler was suffering from severe corrosion and soda-ash failed as a remedy. Examination revealed many blisters on the shell below the water line and each blister contained a liquid which held in suspension a black powder, which on analysis proved to be ferrous sulphide, while the liquid was free sulphuric acid. The feed water used was alkaline and free from sulphur.

### Peculiar Action of Soda-Ash

It is some time since Heyn and Bauer confirmed the fact that diluted solutions of sodium and potassium carbonate promote corrosion, instead of stopping it, but even now very few engineers who use soda-ash know this and consequently do not know whether or not they are using soda-ash to the best advantage. The action of soda-ash is peculiar when used in boilers. It does not seem credible that a diluted alkaline solution will increase in corrosivity up to a certain concentration and then decrease to zero or nearly so, as the solution is made stronger, yet this has been proven true several times within recent years.

### Interesting Experience with Soda-Ash

The most recent and perhaps the most interesting experiments on the action of soda-ash in steam boilers, are those conducted by Lieutenant Commander Frank Lyon, U. S. A.[1]

Lieut. Lyon was in charge of the boilers of the "Oregon," which were of the tubular type and fed with hot water from open tanks; the water being kept highly alkaline. Corrosion aboard the vessel was practically unknown. He later spent three years as senior engineer officer aboard the "New Jersey," the boilers of which were of the B. & W. type. Soda-ash had proven so good aboard the "Oregon" that Lyon used it in the "New Jersey's" boilers, but owing to the drums of these boilers being smaller than those of the "Oregon," the amount of soda-ash used was less because of fear of serious foaming. Corrosion troubles of the worst kind began immediately; in a period of two weeks twelve changes of boilers for renewing pitted tubes were made.

### Weak Soda-Ash Solutions Corrosive

After three years in experimental investigation of the problem of preventing corrosion, Lyon learned what others before him had learned, namely, that different concentrations of soda solutions had different corrosive effects on various metals. He learned that the soda solution used aboard the "New Jersey" was so weak as to be most corrosive. If the solution had been of the strength or concentration of that used aboard the "Oregon" no trouble would have resulted.

Lyon called the concentration whose strength is just below the one in which local corrosion first occurs, the lower *limit* concentration, the one in which the rate of corrosion is maximum, the *critical* concentration, and the one just above that at which the last sign of corrosion appears, the *upper limit* concentration.

### Proper Strength of Solution to Use

Without going into the details of these experiments, we can get the most important information for our purpose from Lyon's statement that "Any boiler using any water can be kept from corroding for any length of time, if treated with soda, and if its concentration is maintained at or above 3 per cent. normal alkaline strength."

The normal solution of soda-ash is 53 grams of pure soda-ash dissolved in 1 liter of water; 3 per cent. normal would be 55 grams per liter or 208 grams per gal. Understand from this that the water in the boiler must contain at least 208 grams of soda-ash per gal. in order to be of a strength of alkalinity that prevents corrosion, but do not suppose that 208 grams or nearly ½ lb. must be fed in with each gallon of water during normal operation of the boiler.

### Initial Charge of Soda-Ash

Suppose we are to put a 100-hp. tubular boiler in service with water charged to 3-per-cent. alkalinity. The boiler will have approximately 5000 lb. or about 600 gal. of water when full to the normal water level. If 208 grams of soda-ash is needed per gallon of water, then we will have to put

$$\frac{208 \times 600}{453.6} = 275.13$$

say 276 lb. of soda-ash into the original water that enters the boiler preparatory to starting up.

Unless the boiler primes or there are water leaks or the blowoff is used frequently, little soda-ash will be lost from the boiler and the solution will remain at the proper concentration. After charging the water it is best, perhaps, to operate the boiler under usual conditions, titrating samples of the water taken from the boiler every few days to ascertain how much the solution has weakened. In this way it will not be long before one can

*From a lecture before the Technical Society of the Brooklyn Union Gas Co.

[1]"Journal" of American Society of Naval Engineers for August, 1912.

feed the proper amount of soda-ash to keep the alkalinity at about 3 per cent. normal and titrating will not be necessary except at long intervals, say two weeks. Those who do not understand titrating could use the old but simple method of suspending a piece of clean boiler tube in a sample of the water and add more soda-ash to the boiler water if the piece of tube shows signs of corroding in a period of 12 hr.

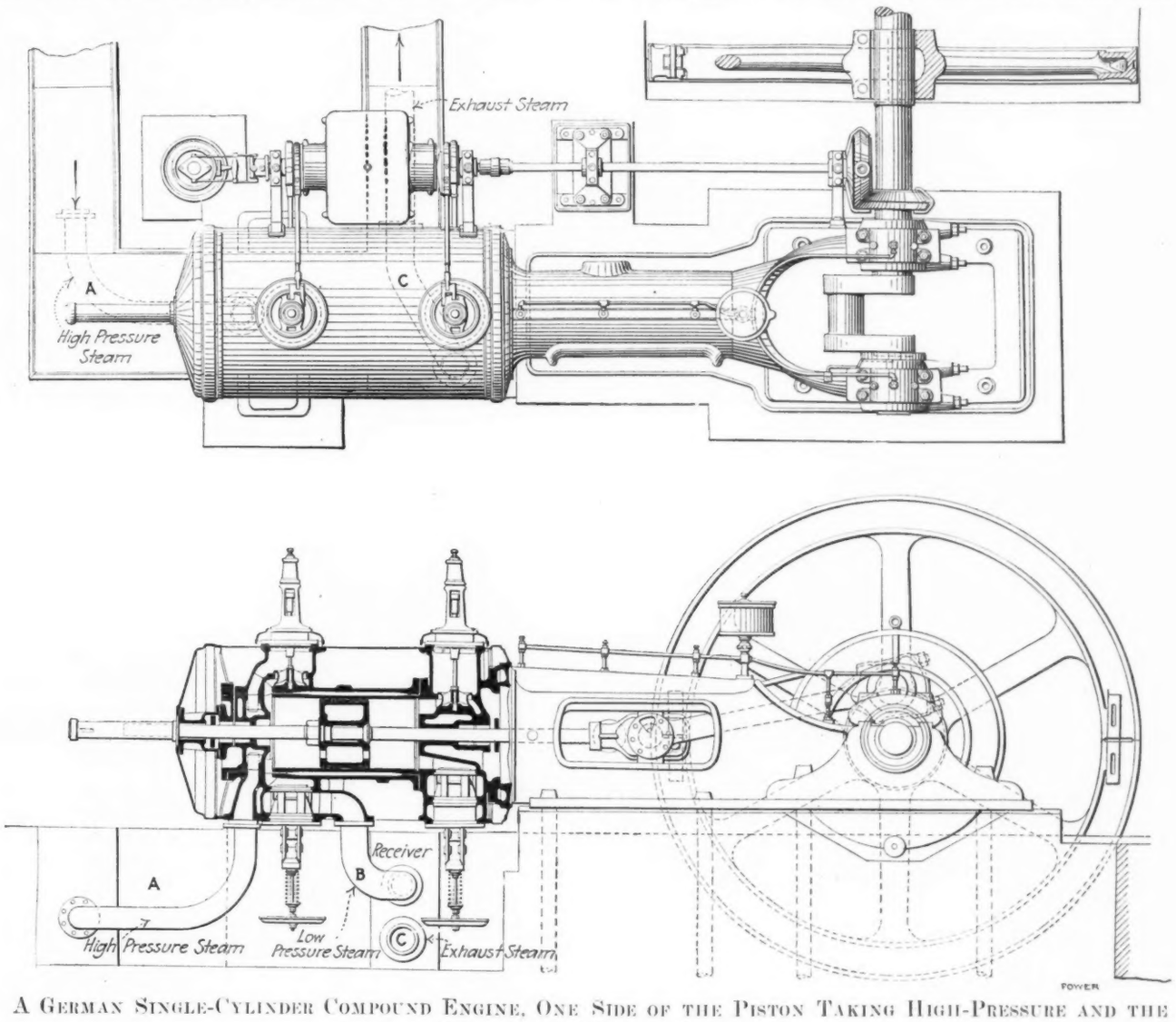
### Boilers Unaffected but Mains Corroded

One of the most serious troubles accompanying the use of soda-ash is the corrosion, often rapid, of the steam main and its valves, though the boiler itself is free from corrosion, and perhaps the main does not corrode near the boiler but is seriously affected at points a short distance therefrom. This is due to priming or wet steam produced by other causes. A "slug" of water of high (safe) concentration goes over into the main, but before being drained away the concentration is reduced to the critical (unsafe) concentration by the water of condensation from the steam which is not alkaline. The best way of overcoming this trouble is to provide ample drains in the main near the boiler so as to get rid of the water before its alkalinity is reduced to the critical concentration.

When boilers prime due to the combined action of soda-ash and the impurities in the water it is well known that sodium phosphate or tannic acid will tend to prevent priming.

## A Single-Cylinder Compound Engine

If a considerable part of the steam exhausted by the high-pressure cylinder of a compound engine is bled from the receiver for heating and manufacturing processes the low-pressure cylinder can be much smaller than if it had to work all of the steam which was admitted to the engine. We recall a compound at the Pacific Mills at Lawrence which was run backwards, that is the high-pressure steam was admitted to the larger, or low-pressure, cylinder, because so much was taken from the receiver that the smaller cylinder was large enough to expand and use what was left.




A German Single-Cylinder Compound Engine, One Side of the Piston Taking High-Pressure and the Other Low-Pressure Steam

It is readily apparent, then, that under favorable conditions of bleeding the two cylinders of a compound engine might be of equal diameter, or rather that so much

of the exhaust of the high-pressure cylinder may be used that the rest of the steam may be worked in a cylinder of the same diameter with the effect of a compound engine.

In the Missong, single-cylinder compound engine, built by Thyssen & Co. A. G. at Mülheim, this idea is utilized by using one end of a cylinder as the high- and the other end of the same cylinder as the low-pressure stage. Steam at boiler pressure comes to the engine through the pipe *A*, and is worked in the head end of the cylinder. The pipe *B* carries the high-pressure exhaust to the receiver, from which as much of the steam as is needed for heating, etc., is taken at *C*. The remainder passes to the crank end of the cylinder, where it is further expanded and exhausted to the condenser. The system is applicable only when at least one-half of the high-pressure exhaust can be used outside of the engine.

## Beam Automatic Smoke Preventer

Another smoke-preventing device has been patented and is designed to automatically regulate the admission of air and steam to the furnace of a boiler. The device consists of a regulator for controlling the smoke-preventing apparatus, and a door deflector which causes the entering air to mingle with the steam from the jets. Fig. 1 shows the device, which is manufactured by the Rockford Smoke Preventing Co., Rockford, Ill., applied to a set of boilers. A detailed view of the apparatus is shown in Fig. 2.

This device consists of a rocker-shaft *A*, Fig. 2, mounted in brackets fastened to the furnace front. The

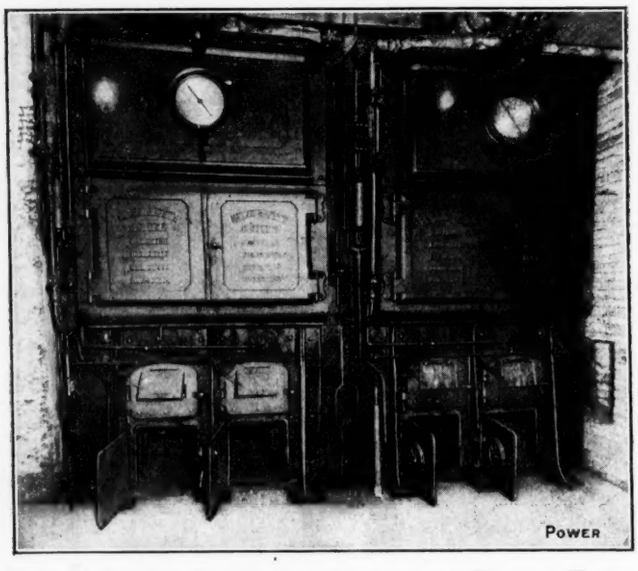



FIG. 1. SMOKE PREVENTER APPLIED TO RETURN-TUBULAR BOILERS

shaft carries the arms *B* which have a chain connection with the doors covering the air opening in the fire doors. This shaft is rocked through a link connection *C*, the arms being actuated through the contact of the fire doors when open, with the contact pieces on the lower shaft. Steam is supplied through the main pipe *D* and a control valve passes through the pipe *E*, from which it discharges through discharge jets over the fire in the furnace.

The regulator operates by the passage of oil through valves and the filling chamber which is filled nearly full with oil. When the fire door is open the motion is transmitted to the rocker-shaft *A*, and the piston in the cylinder is raised through the crosshead connection. This movement operates the damper and increases the draft, opens the steam-control valve, permitting steam to be discharged from the jets into the furnace, and also opens the air doors through which air is drawn and mixed with the steam from the jets.

As the piston rises the oil contained above it passes

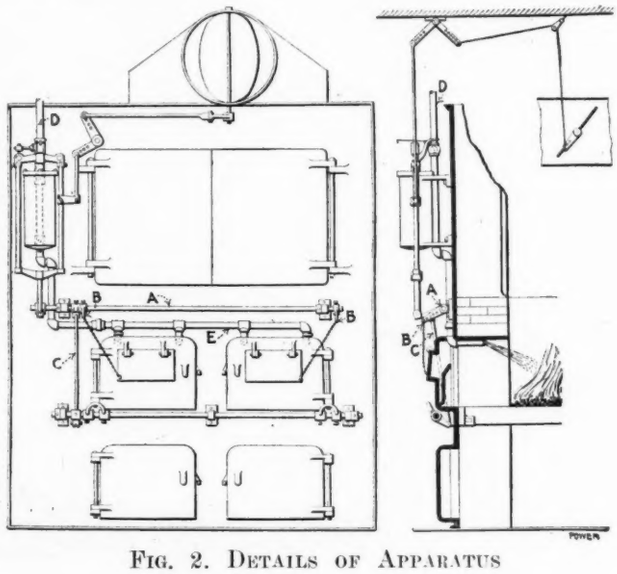



FIG. 2. DETAILS OF APPARATUS

through a valve which is opened by the weight of the oil. When the fire has been charged with fuel and the furnace door is closed, the weight of the operating part causes the piston to descend when the valve will be seated, and the oil will pass upward through a valve, thus causing the piston to slowly descend. During the time the piston is descending, steam and air are being discharged into the furnace, and as the piston reaches the end of its descent the oil in the bottom passes upward through a bypass, giving the piston a final quick movement to shut off the steam supply.

By a screw adjustment the degree of opening of a needle valve may be varied, and the length of the time required for the piston to descend is changed. When the piston is at the upper end of its travel and it is desired to have it descend quickly to an inoperative position, a bypass lever can be pulled, unseating a valve and permitting the oil to pass upward without restriction.

**Illinois Coal Production in 1912**—The production of coal in Illinois in 1912, reached the great total of 59,885,226 short tons, with a value at the mines of $70,294,338. These are record-breaking figures for the state, according to E. W. Parker, of the United States Geological Survey. The production in 1912 exceeded the previous maximum, 53,679,118 tons, made in 1911, by 11.6 per cent. The value increased 18.1 per cent. Illinois still fell below West Virginia in tonnage and continued third in rank among the coal-producing states. With regard to the value of the output, however, Illinois beat West Virginia by more in dollars than it was surpassed by that state in tonnage.

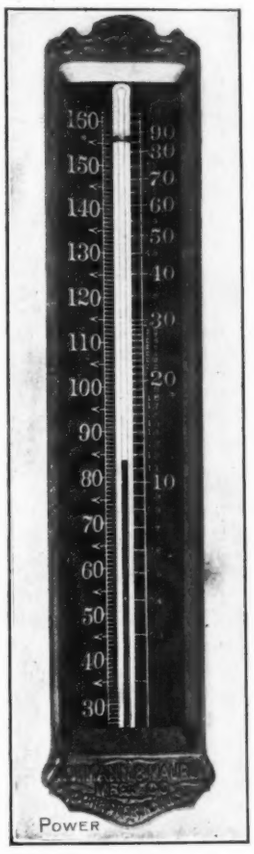
**Texas Coal Production in 1912**—The production of coal in Texas in 1912 passed the 2,000,000-ton mark, with a value at the mines of $3,655,744. These figures are record breakers for the state. The output of lignite in 1912 reached nearly 1,000,000 tons.

# Absolute Pressure Thermometer

By W. H. Phillips

*SYNOPSIS—By the use of a special thermometer the engineer is enabled to read the the temperature of steam in degrees F., and the absolute pressure in inches of the saturated steam in a turbine-exhaust head.*

On steam turbines operating condensing, vacuum readings are usually taken by means of a mercury column, connected to the exhaust head of the turbine. These readings, subtracted from the barometer readings taken simultaneously, give, after making proper corrections for temperature and other variables, the absolute pressure, or back presure, against which the turbine is exhausting. Some engineers and turbine and condenser manufacturers specify or guarantee certain results when a specified number of inches of vacuum are obtained, referred to a 30-in. barometer. This is confusing to many and unnecessarily complicates an otherwise simple matter.




Absolute Pressure Thermometer

Two inches absolute pressure means the same whether obtained at sea level with a barometer at 30 in. or at an elevation of 10,000 ft. with a barometer registering 20 in. This great difference of elevation, of course, affects the amount of available energy in steam of the same initial gage pressure, but the back pressures have the same value and the same temperature. Saturated steam at 1 in. absolute pressure has a temperature of 79 deg. F. regardless of the barometer. No matter what the absolute pressure may be, we can find the temperature corresponding to this pressure in a steam table, and *vice versa*, knowing the temperature, the corresponding pressure can be obtained.

The writer has had a special thermometer made as shown herewith, graduated on the left side in degrees F., and on the right in inches and fractions of an inch absolute pressure. The two graduations are so arranged as to enable one to read the absolute pressure of saturated steam corresponding to the temperature recorded directly opposite. The cut shows the thermometer registering 84 deg. On the right-hand scale we read 1.16 in. absolute, which, if the instrument were connected to a turbine-exhaust head, would indicate that the turbine was exhausting against 1.16 in. back pressure or within that amount of a perfect vacuum.

This instrument was connected to the exhaust head of a 3500-kw. turbine located in a power station where readings are taken every hour of each of three watches. The operators were instructed to take the absolute pressure reading. All other readings were taken off the station instruments. Those of interest in this test consisted of a Standard U. S. Weather Bureau mercurial barometer and Standard H. & M. mercury column, both located on a gage-board within 6 ft. of the temperature pressure-gage. The mercury-column connection was tapped into the exhaust head of the turbine within 8 in. of the special thermometer being tested. All readings were taken under ordinary operating conditions and by the three regular operators on their different watches. The test lasted five days, the load on the turbine varying from 1200 to 3900 kw. and the vacuum from 1.3 to 2.4 in. absolute. Every reading taken was used in making the average result as given herewith:

| | |
|---|---|
| Average absolute pressure as taken on mercury column and barometer, corrected for temperature | 1.941 in. Hg. |
| Average absolute pressure taken on special thermometer | 1.935 in. Hg. |
| Average difference in 100 readings | 0.006 in. Hg. |

The greatest difference found between these two methods was when the turbine was carrying an overload, and only under this condition was the temperature-absolute-gage reading higher than that taken by the mercury column and the barometer. The greatest difference, 0.110 in., being at 3900 kw., showing that the increased volume of steam was slightly superheated in passing through the last stages of the turbine.

The only conditions which may interfere with the accuracy of these readings are, the presence of air, conduction through the metal case of the instrument and superheat. Regarding air, Prof. Charles K. Carpenter, (on page 681, May 13, 1913) writes as follows: "Most of the air in the condenser seeps in through the pores of the condenser and the exhaust piping, and their joints, particularly the latter, the percentage of air present in the steam at the exhaust nozzle of a steam turbine is negligible." This statement agrees with the writer's experience that any leakage in the turbine nozzle head is usually at the shaft packing, which is easily discovered and corrected.

The inaccuracy due to conduction has been found to be negligible in well constructed thermometers properly installed. Superheat at the exhaust of a well designed turbine is impossible, except under heavy overload conditions.

The object of this article is not to advocate the use of the thermometer absolute gage for test purposes, but for station readings it is sufficiently accurate, easily read, and the chances of errors are minimized.

## Power-Plant Records in a Small Station

*SYNOPSIS—Records of the plant were kept by means of a daily log and a card system. The latter provided a means for checking the maintenance of the plant.*

Several years ago, soon after taking charge of a pumping station in a Southern city of some 20,000 inhabitants, one of the company's officials asked the engineer if it

were practicable to keep a record of the engine-room expenses.

A daily log, covering such items as were then deemed necessary, thereafter was sent weekly to the office until the company sold out to the municipality. After this no records were sent in, as the city officials did not have the forms printed. Lately, the weekly report has been resumed.

During the interval, however, the records were regularly kept by the engineers and were extended from time to time to cover more details of plant performance. Although it gave much useful information, it did not furnish the information wanted, and the engineer, for his own personal use, adopted a card system which is quite valuable in checking the maintenance of the plant. The system carries not only all items that appear on the log, but many others that it would be impracticable to cover in this way.

All of the apparatus is individualized. In the boiler room, for instance, each boiler is numbered and all the accessory parts have the same number, such as furnace, setting, water column, blowoff valves, safety valve, gages, steam pipes, etc. These cards contain all data relating to the different parts, giving the time, material and nature of the repair. They are then grouped together in the file, all cards of the same heading, such as water column, going together in this specific information of any particular piece of apparatus.

The same order is carried out in the engine room, but in addition to the repair card, each engine and pump has two packing cards, one for steam, the other for water, and a valve card. They have a special value, as they not only furnish information as to the total cost of keeping up the supplies, but are a valuable means of comparing the relative merits of different manufacturers' goods when running under like conditions of service. These cards are filed in a vertical cabinet so arranged that any heading is easily visible. It is always at hand and it is not necessary to look back through a mass of records when information on any particular subject is wanted.

There is no record of the feed water as several makes of meters have been tried from time to time and discarded. Two tanks are erected on platform scales and once or twice a week the water is weighed for several hours, and, as the load is fairly constant, an approximate estimate is made of the average daily water evaporated.

A fireman efficiency record is kept, based on the percentage of $CO_2$ in his flue gases, both on the log and on a monthly form. In addition to the log and card system a loose-leaf ledger is kept, in which the unit cost of water pumped is recorded (the pumpage being based on plunger displacement) consisting of operating, maintenance and fixed charges. As it is not practicable to separate the maintenance and operating charges—the plant is too small to have a special card for upkeep work—the former is divided into five heads: labor, fuel, waste, miscellaneous supplies, and fixed charges. The total cost of each of these items for the 24 hours divided by the number of thousand gallons of water pumped gives the unit charge under that head. The whole combined give the total unit charge for the day.

Coal is the item subject to most variation in making up the daily charge, as it is necessary to estimate the cost at the furnace. This can be arrived at approximately in advance, the coal being weighed as it is hauled from the car and again at the boiler. It might seem that this means a good deal of bookkeeping, but as a matter of fact 30 minutes a day is ample time to check up and make all the necessary computations.

It does not make so much difference how much of a record the engineer tries to keep, it is the regularity with which it is done that counts. The daily log may be filled out in the evening, but the card system should be attended to immediately after making a repair, for if the charge is forgotten the system is valueless. The engineer may not have all the apparatus for accurately determining the output or performance of his plant, but he can know a great deal about it.

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## Public Service Commission Acts in Landlord and Tenant Case

An important decision has been made in the case of Alonzo B. Knight against the United Electric Light & Power Co., of New York, in which the Public Service Commission has ordered certain contract riders to the rate schedules discontinued. These riders were as follows:

INCLUSION OF TENANTS' CONSUMPTION—WHOLESALE RATE

It is further understood and agreed that the customer will exclude all other electric service from the building and not install a private plant for light and power during the term of this contract, and to guarantee the use of 100,000 kw.-hr. annually in said building. Said guarantee shall include all current consumed by the tenants in the building as well as all current consumed by the undersigned. It is understood, however, that the accounts of the tenants shall have no further connection with the contract.

CONJUNCTIONAL SERVICE

It is further understood that in view of the fact that the buildings enumerated are supplied under the wholesale contract, are not more than 100 ft. apart, are under common ownership or leasehold, and may be served from a single center of distribution, the current required for them may be taken collectively in determining the rate to which the undersigned is entitled under this contract.

INCLUSION OF TENANTS' CONSUMPTION—GENERAL RATE

It is further understood and agreed that the owner or buildings enumerated in this contract are under common ownership or leasehold, are intercommunicating within the building, and are classed under one fire risk and operated as a single property, and that the owner agrees to purchase under this contract the current for all the apartments within the building or buildings, that the United Electric Light & Power Co. will furnish and set meters for said apartments and will render if required by the customer an itemized statement under this contract, giving tne readings and numbers of the individual meters. Stores within the building or buildings are not classed as apartments. They may be included at the option of the owner or lessee.

It is further understood and agreed that the building or lessee will be responsible for the consumption of current within the building included herewith.

According to the first two riders the tenants under the general rate would pay 10c. per kw.-hr. for the first 250 kw.-hr. of monthly consumption, the rate gradually decreasing until it becomes 5c. per kw.-hr. for an excess of 1500 kw.-hr. of monthly consumption, the wholesale rate being available only upon the guarantee of 100,000 kw.-hr. of annual consumption. The landlord, on the other hand, in consideration of his promise to exclude all other service and not install a private plant, is granted a rate based not upon his own consumption but upon that of all his tenants taken collectively. In a memorandum

accompanying the order, Commissioner Maltbie cites the following example:

Assume that the landlord uses in one year 3000 kw.-hr. and in the month in question 200 kw.-hr.; that the total consumption of landlord and tenants is 150,000 kw.-hr. annually and that a certain tenant's bill for the month is for 200 kw.-hr.

The landlord will pay an average of 4⅗c. per kw.-hr. or $9.30, whereas the tenant will pay at the rate of 10c. per kw.-hr. or $20 for the same amount of electric energy.

The second rider simply permits the above to be applied collectively to all buildings under common ownership or leasehold which are not more than 100 ft. apart and which can be served from a single center of distribution.

Under the third rider the electric company installs and reads the meters on the tenants' premises, rendering to the landlord a statement accordingly. The landlord pays for this current at the wholesale rate and in turn charges the tenants at the retail rate, the difference amounting to a handsome profit for the landlord. An example under this rider cited by Commissioner Maltbie is as follows:

Assume an owner or lessee consumes in a certain month 5000 kw.-hr. and that no one of them uses more than 250 kw.-hr. The landlord will receive a statement from the company giving the reading of each meter and the total, which in the case assumed will be computed as follows:

| | | |
|---|---|---|
| First | 250 kw.-hr. at 10c. | $25.00 |
| Next | 250 kw.-hr. at 9c. | 22.50 |
| Next | 250 kw.-hr. at 8c. | 20.00 |
| Next | 250 kw.-hr. at 7c. | 17.50 |
| Next | 500 kw.-hr. at 6c. | 30.00 |
| Next | 3500 kw.-hr. at 5c. | 175.00 |
| Total | 5000 kw.-hr. at 5.8c. | $290.00 |

The landlord may now charge the tenants at the regular 10c. rate, which will be

4800 kw.-hr. at 10c. = $480

From this it appears that the landlord will get his current free and in addition will make a profit of $190 for a single month. But as he would have to pay at the 10c. rate if he had no rider, the latter has improved his condition to the extent of $210 per month as compared with the landlord who has no such rider in his contract. In a year this would amount to a fine profit.

## Boiler Manufacturers' Convention

The twenty-fifth annual convention of the American Boiler Manufacturers' Association was held in the Hollenden Hotel, Cleveland, Ohio, on Sept. 1 to 4, inclusive. As usual, uniform boiler specification laws were the chief matter to receive attention.

The first session was opened at 2 o'clock on Monday afternoon with the customary address of welcome by the mayor, the Hon. Newton D. Baker, to which Col. E. D. Meier, president of the association responded. A. V. Canfion then gave an address on "Uniform Boiler Specifications" in which he took up the problems encountered and discussed them from their legal aspect.

At the Tuesday morning session the report of the Committee on Uniform Specifications and Construction of Boilers was read by Chairman Meier and approved by the vote of the association. A large delegation of the National Association of Tubular Boiler Manufacturers was present by invitation to listen to the reading of the report and discuss its provisions. The report is a continuation of the one presented last year at the New Orleans meeting and reported in our issue of April 2, 1912. During the year the committee send out a large number of letters requesting criticisms and comments on the report as made last year. From the 17 replies all of which were favorable it was assumed the report required no marked alteration. One change recommended was that all plates be stamped with the figure of the even 100 lb. nearest to the tensile strength figure shown by the test piece.

The membership of the National Association of Tubular Boiler Manufacturers, as its name implies, is composed solely of fire-tube boiler makers. It was organized a little over a year ago in Cincinnati and usually meets there about once a month. At the invitation of the American Boiler Manufacturers' Association it held its September meeting in Cleveland during the latter's convention so that the report of the committee on uniform specifications might receive joint consideration. The Tubular Boiler Manufacturers' Association has issued a set of standard specifications of its own which has been officially accepted and is used by 34 manufacturing concerns. This will be published in an early issue.

At the Tuesday afternoon session, L. I. Yeomans read a paper on "Factory Efficiency"; W. C. Connelly presented one on "Shop Costs," and T. E. Durban gave a third, on "Uniform Boiler Specifications."

### UNIFORM BOILER SPECIFICATIONS

By way of introduction Mr. Durban stated that he thought that the manufacturers themselves had in a degree obstructed the securing of a uniform specification through petty jealousies and individual anxiety to secure slight selling advantages through changes in specifications. However, with ever multiplying state and city provisions for inspection, etc., and with the advantages and benefits to all of a uniform specification growing more apparent, the desired end seems encouragingly near.

Notwithstanding an existing federal law applying to locomotive boilers for interstate traffic, it was the opinion of President Taft and those senators and congressmen consulted and it is the present opinion of many able lawyers that a national law applying to stationary boilers would not be valid. The fact that a boiler is built in one state and shipped for use in another does not seem to bring it into the class of apparatus controllable by interstate commerce laws. Also, the common-sense argument that whereas the United States provides for marine boiler control it should also provide for stationary boilers, on the ground that life is no less sweet on land than on water, does not carry weight when it comes to legal validity.

Hope in the direction of a national law being faint, it remains for manufacturers and others interested to do all they can toward securing uniformity in the laws passed by the various states and cities. Existing laws have been found to be of great benefit as far as safety of life and property is concerned. But the lack of uniformity in them is a needless hardship on manufacturers and a real expense to the purchaser who, in the end, must pay for the cost of maintaining on hand the large assortment of stock which the manufacturer is forced to carry so that he may satisfy the diverse conditions imposed by the laws of various localities.

One glaring abuse, resulting from imperfectly framed laws, exists because in some cases the steel for some uses is specified as firebox steel while in other instances flange steel is satisfactory. Mr. Durban stated that from data in his possession apparently there was practically no difference between firebox and flange steel that came from the same mill as both showed the same chemical and physical properties. Therefore, it is his opinion that the difference between flange and firebox steel to pass Massachusetts, Ohio and Detroit inspection and the ordinary steel consists mainly of the extra charge which the mill must make in order to put the required number of stamps on the plate to meet the law. What Mr. Durban considers to be of far greater importance than stamps showing brand names such as firebox, flange, etc., are stamps showing mill-test numbers. These should be so placed that they would not become lost in the process of manufacture and would be accessible to the inspector.

In summing up, these specific recommendations were made: A factor of safety of 5. Specification of a uniform steel and that this specification cover flange steel. That the steel be marked with heat numbers instead of the brand name. A minimum tensile strength of 60,000 lb. per sq.in. That the steel be subject to the bending and quenching test. That the phosphorus content must not exceed 0.03 per cent.; sulphur, 0.04 per cent., and manganese, 0.5 per cent. That weldless crowfoot type braces be used of the same quality of steel as the plate, and that they be figured on a basis of 7500 lb. per sq.in. of section in the brace, and that on this basis no brace be used containing more than 1.28 sq.in. of sectional area. That through braces either above or below the tubes be weldless and be figured on 7500 lb. per sq.in. of area; that all seams be butt strapped with inside and outside covering strips. That all manholes be 11x15 in. That no plates be used in a tubular boiler thicker than ⅝ in. That all holes for rivets be punched ¼ in. small and reamed to size. That all flue holes be punched ½ in. small and reamed to size. That no cast iron be used in connection with the boiler either for reinforcement or any other purpose.

That all plates be beveled on a planer instead of sheared on a bevel shear. That water column connections all be 1¼ in.

Demonstration on a $\frac{3}{4}$-in. plate showed that it is impossible to do a good job on a splitting shear, and that the same objection that exists to a punched hole, holds good in the use of a splitting shear on thick plates. That is, that the metal is distorted by use of a splitting shear.

At the annual election on Thursday the following officers were elected: E. D. Meier, New York City, president; J. D. Farasey, Cleveland, Ohio, secretary; Joseph F. Wangler, St. Louis, Mo., treasurer; Thomas M. Rees, Pittsburgh, Penn., first vice-president; J. Don Smith, Charleston, S. C., second vice-president; H. D. MacKinnon, Bay City, Mich., third vice-president; L. E. Connelly, Cleveland, Ohio, fourth vice-president, and Robert Jay, Oswego, N. Y., fifth vice-president. The first six officers mentioned were reëlected, according to customary proceedure. New York City was chosen as the place for the next annual meeting which will be held in September, 1914.

Prominent among the entertainment features was an excursion by boat to Cedar Point which occupied the entire day on Wednesday and the annual banquet Thursday evening. The latter event was an especially festive one as it commemorated the twenty-fifth anniversary of the organization. Other entertainment features were an informal reception on Monday evening, an automobile ride for the ladies on Tuesday, a theater party on Tuesday evening and a shopping tour for the ladies on Thursday.

## Boiler Carries 318 Per Cent. of Rating

In Bulletin 51, the Edge Moor Iron Co. gives results of three interesting tests on one of its 736-hp., water-tube boilers equipped with a Foster superheater and a Taylor stoker. The tests were made at the Westport generating station of the Consolidated Gas, Electric Light & Power Co., of Baltimore. The boiler contained a total of 7361 sq.ft. of water-heating surface and 429 sq.ft. of superheating surface.

In the first test of two hours' duration, 2340 boiler horsepower was developed. This is 318 per cent. of the rated capacity. The other two tests were run for eight hours and the horsepower developed was 210 and 248 per cent. of rating, respectively. In the order given the evaporation per square foot of water-heating surface was 10.97, 7.26 and 8.57 lb. The steam pressures were 172.1, 169.7 and 167.4 lb. In the same order the steam was superheated 103.5, 130.4 and 106.6 deg. The latter part of the bulletin is devoted to the time required to obtain a high rate of evaporation from a banked fire, and to the value of rear passes in a boiler.

## A Cold-Storage Banquet

Foreign delegates to the Third International Refrigeration Congress which takes place in Chicago from Sept. 17 to Oct. 1 will be the guests of honor at a unique banquet at which only foods and beverages that have passed through refrigeration will be served. Most of the foodstuffs have been collected by the United States Department of Agriculture. Each delegate at the feast will be presented with a souvenir menu giving an officially certified history of each article of food served.

The frozen eatables now at Washington will be sent to Chicago prior to the opening of the refrigeration show to be held at the International Amphitheater and will be in the huge $10,000 ice box which will occupy the most conspicuous space at the exposition and will be used exclusively for the government exhibits.

Organizations which have arranged to hold annual or special meetings during the congress include the American Meat Packers' Association, American Association of Refrigeration, American Society of Refrigerating Engineers, National Ice Association, Eastern Ice Association, Western Ice Manufacturers' Association, Southern Ice Association, Southwest Ice Manufacturers' Association, Illinois Ice Dealers' Association, Indiana Ice Dealers' Association and the Louisiana Ice Manufacturers' Association. Many other associations will be represented by delegates.

**Fastest Ship in the World**—According to a special cable dispatch to the New York "Sun," the new Russian destroyer "Novik" is the fastest ship in the world, having developed 37.3 knots in her trials at Swanemuende. She was built at that place from designs by the Vulcan Company and was fitted with turbines and boilers made by that company at Hamburg and Stettin.

## NEW PUBLICATIONS

THE GAS ENGINE HANDBOOK, Seventh Edition. By E. W. Roberts. Published by the Gas Engine Publishing Co., Cincinnati, Ohio; 323 pages; 4¼x7 in.; illustrated. Flexible leather. Price, $2.

In order to bring the book thoroughly uptodate much of the text of the previous editions has been rewritten and the formulas on design have been revised, wherever necessary, to correspond with present practice.

The text has been grouped under three general headings, namely: Descriptive Matter, Design, and Operation, Testing and Selection. Under the first heading are taken up the gas-engine cycles, fuels, valves, carbureters, vaporizers and mixers, igniters, governors, automobile, marine and aëronautical motors and lubrication. Under design, are considered the proportioning of the various parts, in which connection the various formulas are given. The third division includes suggestions for selecting an engine to meet different conditions, starting and stopping engines, troubles and how to remedy them, and rules for making and working up gas-engine tests.

The author has made a strong plea for the two-stroke-cycle engine, pointing out the false ideas about this type and the cause of its unpopularity in certain lines.

The whole treatment of the subject is thoroughly practical and the style is clear and concise. We have no hesitancy in recommending it to the engineer or designer desiring a ready reference on gas-engine matters.

## PERSONALS

Freeman Wines, formerly of Newark, N. J., has taken the position of chief engineer of the New Metropolitan Hotel, Asbury Park, N. J.

John A. Stevens, Lowell, Mass., has just completed a term of six years as a member of the Massachusetts Board of Boiler Rules and has been succeeded by Frederick A. Wallace.

Stephen A. Staege and Raleigh H. Bennett, engineers, and Charles E. Dewey, architect, Watertown, N. Y., formed a partnership, Aug. 31, 1913, under the firm name Staege, Bennett & Dewey, engineers and architects.

William H. Sargeant, for the past 13 years chief operating engineer at the plant of the Prudential Insurance Co., Newark, N. J., has been appointed chief engineer. He will now handle the executive as well as the operating end of the plant.

Walter Pryor, for a number of years chief engineer of the Henry R. Worthington plant at Harrison, N. J., now holds the position of superintendent of plants for the Eclipse Tanning Co., with headquarters in Newark, N. J. Mr. Pryor has nine plants under his charge.

Otis Funderburk, formerly Michigan manager for the Findeisen & Kropf Manufacturing Co., maker of the Rayfield carburetor, has been appointed sales manager of the carburetor division of the Detroit Lubricator Company, which is making and marketing the Stewart carburetor.

Frank S. Washburn, of Nashville, Tenn., president of the American Cyanamid Co., Niagara Falls, N. Y., has been elected to fill the unexpired term of James Mitchell, formerly president of the Alabama Power Co., Birmingham, Ala. J. W. Worthington, of Sheffield, Ala., has been chosen one of the vice-presidents. Mr. Washburn is also president of the Goodman Manufacturing Co., of Chicago; is the consulting engineer of the Crescent Coal & Coke Co., Nashville, Tenn.; is a director in the American National Bank of Nashville, and is connected with other industrial and banking institutions.

**Kansas Coal Production in 1912**—The production of coal in Kansas in 1912 increased from 6,178,728 short tons, valued at $9,472,572, in 1911, to 6,986,182 tons, valued at $11,324,130, according to figures compiled by E. W. Parker, of the United States Geological Survey, in coöperation with the Geological Survey of Kansas. The gain amounted to 13 per cent. in quantity, and 19.5 per cent. in value. The average price per ton advanced from $1.50 to $1.62. The increased production in Kansas, as in the other Southwestern States in 1912, may be attributed to the diminished supply of fuel oil and natural gas from the Mid-Continent field. These fuels, especially fuel oil (on account largely of the increased demand for gasoline), have been virtually removed from the steam trade and steam users have returned to coal.